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(54) **SINGLE SPEED AND TWO-SPEED
DISCONNECTING AXLE ARRANGEMENTS**

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2023/0841; B60K 2023/0866; F16H 37/08;
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F16H 48/385; F16H 2048/085; F16H
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See application file for complete search history.

(56) **References Cited**

U.S. PATENT DOCUMENTS

1,362,361 A 12/1920 Starr
2,783,661 A 3/1957 Dryer

(Continued)

FOREIGN PATENT DOCUMENTS

EP 0865954 A2 9/1998
EP 1466777 A2 10/2004

(Continued)

OTHER PUBLICATIONS

International Search Report and Written Opinion for International
Application No. PCT/US2009/053514, dated Mar. 29, 2010. cited by
other.

(Continued)

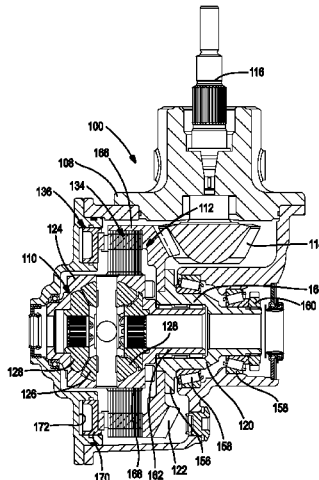
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(57) **ABSTRACT**

A driveline component with a housing, a pinion mounted in
the housing for rotation about a first axis, a ring gear and a pair
of output members. The ring gear is meshed with the pinion
and is rotatable about a second axis. The ring gear is sup-
ported for rotation relative to the housing by a pair of tapered
roller bearings that are disposed along the second axis on a
common lateral side of the ring gear. The output members are
drivingly coupled to the ring gear.

14 Claims, 17 Drawing Sheets



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2023/085 (2013.01)
- (56) **References Cited**
- U.S. PATENT DOCUMENTS
- | | | | | | | | |
|-----------|------|---------|----------------------|--------------|------|---------|--------------------------------|
| 3,749,217 | A | 7/1973 | Bush et al. | 6,533,090 | B2 | 3/2003 | Osborn et al. |
| 4,381,828 | A | 5/1983 | Lunn et al. | 6,547,025 | B1 | 4/2003 | Gassmann et al. |
| 4,407,387 | A | 10/1983 | Lindbert | 6,557,677 | B2 | 5/2003 | Peura |
| 4,480,500 | A * | 11/1984 | Yamamori et al. | 6,592,487 | B2 | 7/2003 | Gassmann |
| 4,763,748 | A * | 8/1988 | Miura et al. | 6,769,506 | B2 | 8/2004 | Gassmann et al. |
| 4,763,749 | A | 8/1988 | Miura et al. | 6,779,420 | B2 | 8/2004 | Peura |
| 4,774,857 | A | 10/1988 | Heine et al. | 6,805,653 | B2 | 10/2004 | Krzesicki et al. |
| 4,782,721 | A | 11/1988 | Dick | 6,814,682 | B2 | 11/2004 | Spitale |
| 4,875,978 | A | 10/1989 | Hiketa | 6,827,663 | B2 | 12/2004 | Tucker-Peake |
| 4,915,190 | A | 4/1990 | Iwata | 6,849,017 | B2 | 2/2005 | Nett |
| 4,938,306 | A | 7/1990 | Sumiyoshi et al. | 6,851,501 | B2 | 2/2005 | Gassmann |
| 5,041,069 | A | 8/1991 | Horst | 6,945,899 | B2 | 9/2005 | Peura |
| 5,061,229 | A | 10/1991 | Tsukamoto et al. | 6,974,400 | B2 | 12/2005 | Williams |
| 5,065,639 | A | 11/1991 | Flanhardt et al. | 7,094,172 | B2 | 8/2006 | Ishikawa |
| 5,069,305 | A | 12/1991 | Kobayashi | 7,150,694 | B2 | 12/2006 | Mizon et al. |
| 5,083,986 | A | 1/1992 | Teraoka et al. | 7,309,301 | B2 | 12/2007 | Janson et al. |
| 5,105,901 | A | 4/1992 | Watanabe et al. | 7,331,896 | B1 | 2/2008 | Kroppe |
| 5,105,902 | A | 4/1992 | Wilson et al. | 7,416,505 | B2 | 8/2008 | Downs |
| 5,156,247 | A | 10/1992 | Wiese et al. | 7,452,301 | B2 | 11/2008 | Yoshioka |
| 5,161,636 | A * | 11/1992 | Haupt et al. | 7,533,754 | B2 | 5/2009 | Burrows et al. |
| 5,174,408 | A | 12/1992 | Hock | 8,042,642 | B2 | 10/2011 | Marsh et al. |
| 5,188,194 | A | 2/1993 | Gasch | 8,047,323 | B2 | 11/2011 | Downs et al. |
| 5,234,072 | A | 8/1993 | Chludek | 8,215,440 | B2 | 7/2012 | Hoffmann et al. |
| 5,314,039 | A | 5/1994 | Hock | 8,469,854 | B1 | 6/2013 | Downs et al. |
| 5,411,110 | A | 5/1995 | Wilson et al. | 2002/0032096 | A1 | 3/2002 | Gassmann |
| 5,423,235 | A | 6/1995 | Botterill et al. | 2003/0040394 | A1 | 2/2003 | Palazzolo |
| 5,484,033 | A | 1/1996 | Frank et al. | 2003/0089185 | A1 | 5/2003 | Hock et al. |
| 5,547,430 | A | 8/1996 | Gasch | 2004/0198548 | A1 | 10/2004 | Showalter et al. |
| 5,562,566 | A | 10/1996 | Yang | 2005/0023063 | A1 | 2/2005 | Mueller |
| 5,599,249 | A | 2/1997 | Zalewski et al. | 2006/0014602 | A1 * | 1/2006 | Sayama 475/231 |
| 5,632,185 | A | 5/1997 | Gassmann | 2006/0283654 | A1 | 12/2006 | Krisher |
| 5,662,543 | A | 9/1997 | Forsyth | 2007/0289797 | A1 | 12/2007 | Bowen |
| 5,690,201 | A | 11/1997 | Gassmann | 2008/0128234 | A1 | 6/2008 | Mogami et al. |
| 5,706,923 | A | 1/1998 | Gassmann | 2008/0227582 | A1 | 9/2008 | Peura |
| 5,762,578 | A | 6/1998 | Forsyth | 2009/0160274 | A1 | 6/2009 | Aikawa et al. |
| 5,839,986 | A | 11/1998 | Yamazaki | 2010/0038164 | A1 | 2/2010 | Downs et al. |
| 5,935,036 | A | 8/1999 | Gassmann et al. | 2010/0044138 | A1 | 2/2010 | Marsh et al. |
| 5,951,428 | A | 9/1999 | Itoh et al. | 2010/0216593 | A1 | 8/2010 | Ekonen et al. |
| 5,980,415 | A | 11/1999 | Showalter | 2011/0275470 | A1 | 11/2011 | Ekonen et al. |
| 6,113,512 | A | 9/2000 | Williams | 2011/0319213 | A1 | 12/2011 | Ekonen et al. |
| 6,116,392 | A | 9/2000 | Gassmann | 2012/0011952 | A1 * | 1/2012 | Schwekutsch et al. 74/333 |
| 6,186,258 | B1 | 2/2001 | Deuschel et al. | 2012/0029779 | A1 | 2/2012 | Dickinson et al. |
| 6,209,673 | B1 | 4/2001 | Barlage et al. | 2012/0073929 | A1 | 3/2012 | Grutter et al. |
| 6,241,067 | B1 | 6/2001 | Hock | 2012/0083380 | A1 | 4/2012 | Reed et al. |
| 6,263,995 | B1 | 7/2001 | Watson et al. | 2012/0204664 | A1 * | 8/2012 | Peura et al. 74/405 |
| 6,267,214 | B1 | 7/2001 | Kwoka | 2012/0234120 | A1 * | 9/2012 | Fukuda et al. 74/405 |
| 6,431,337 | B1 | 8/2002 | Hock et al. | 2013/0055833 | A1 * | 3/2013 | Fukuda 74/15.86 |
| 6,443,282 | B1 | 9/2002 | Kwoka | 2013/0303326 | A1 | 11/2013 | Downs et al. |
| 6,446,773 | B2 | 9/2002 | Kwoka | | | | |
| 6,461,267 | B1 | 10/2002 | Paielli | | | | |
| 6,485,389 | B1 * | 11/2002 | Bell et al. | | | | |
| 6,520,885 | B2 | 2/2003 | Gassmann et al. | | | | |
- FOREIGN PATENT DOCUMENTS
- | | | | |
|----|---------------|----|---------|
| KR | 2019990020807 | U | 6/1999 |
| KR | 100274035 | B1 | 12/2000 |
| KR | 100461078 | B1 | 12/2004 |
| KR | 100483163 | B1 | 4/2005 |
| KR | 20100078173 | A | 7/2010 |
| WO | WO-03-038296 | A2 | 5/2003 |
| WO | WO-2010104853 | A2 | 9/2010 |
- OTHER PUBLICATIONS
- International Preliminary Report on Patentability (Chapter I of the Patent Cooperation Treaty) for PCT/US2009/053514, issued Feb. 15, 2011, cited by other.
- International Search Report and Written Opinion for International Application No. PCT/US2010/041767, dated Feb. 28, 2011, cited by other.
- * cited by examiner

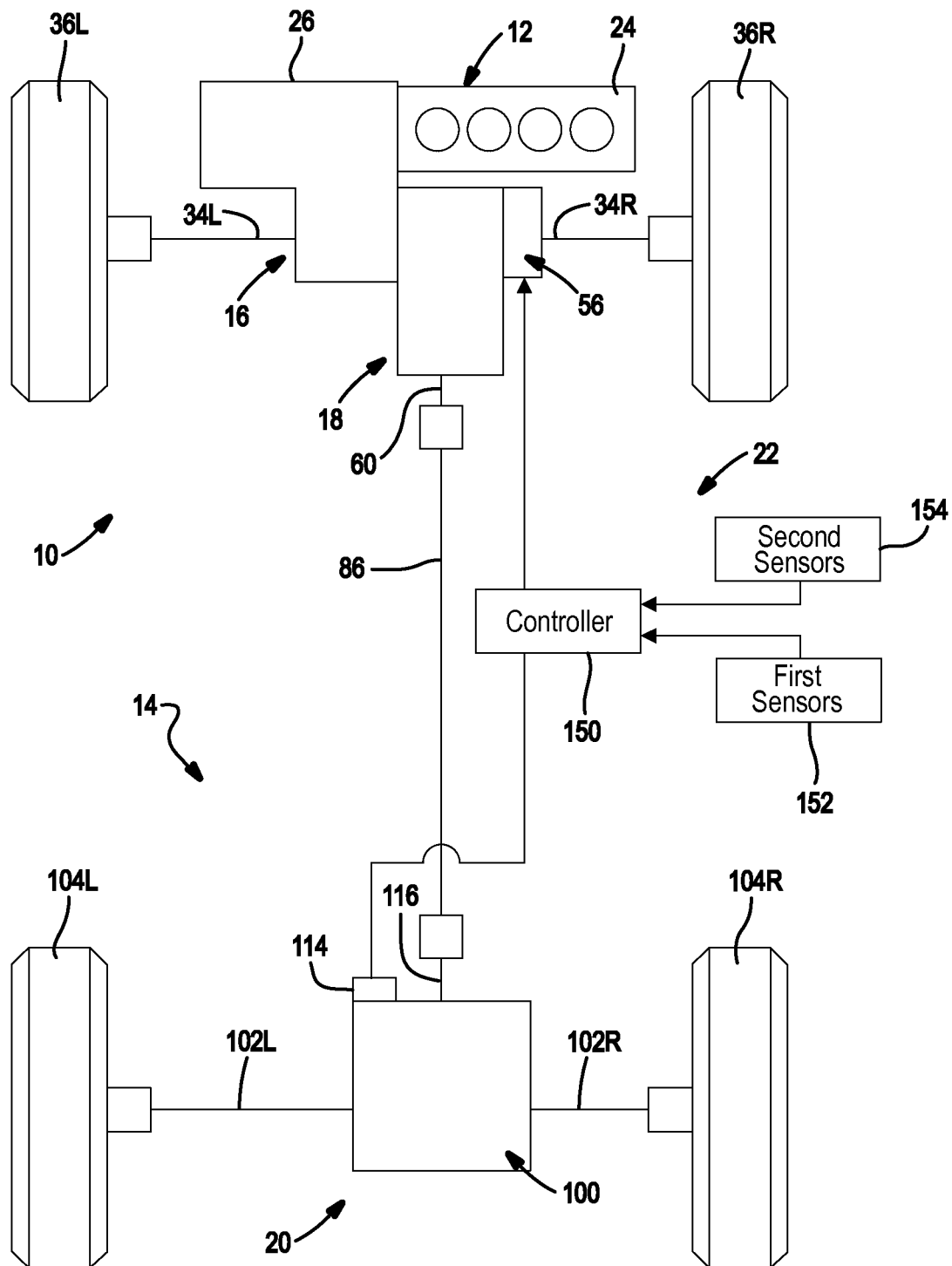


FIG. 1

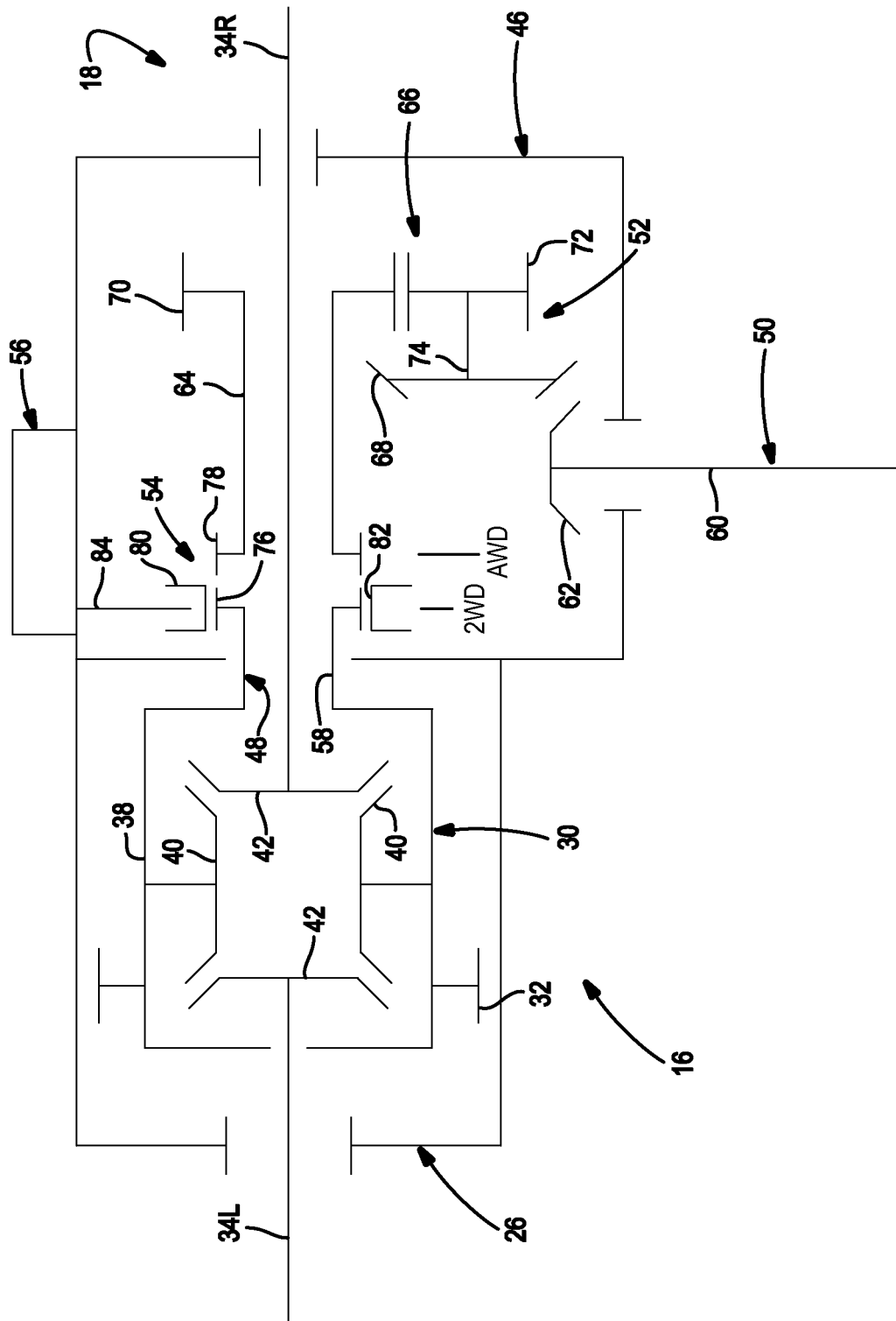


FIG. 2

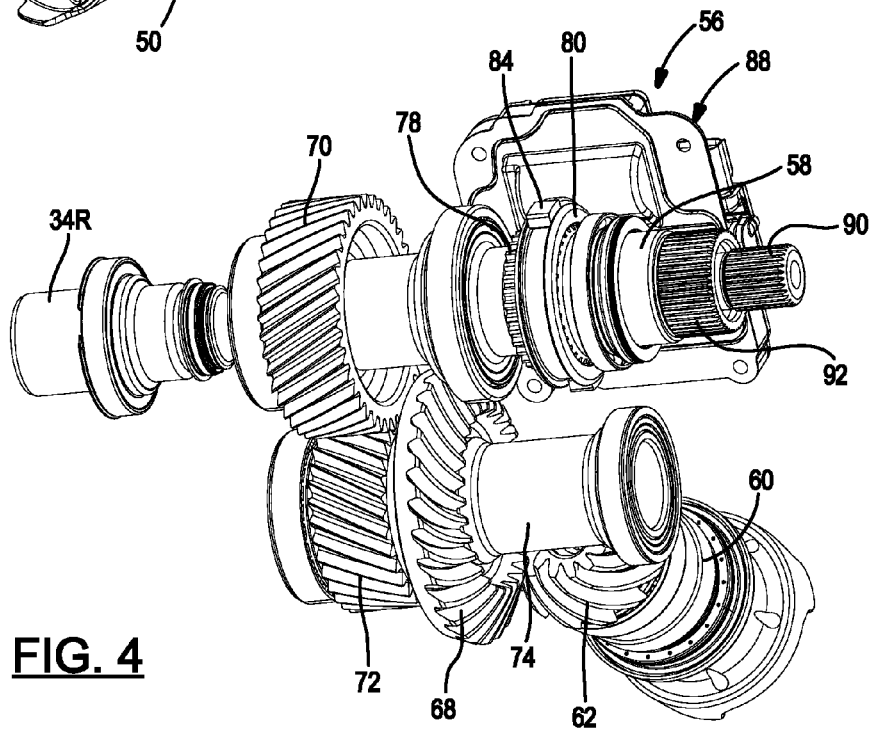
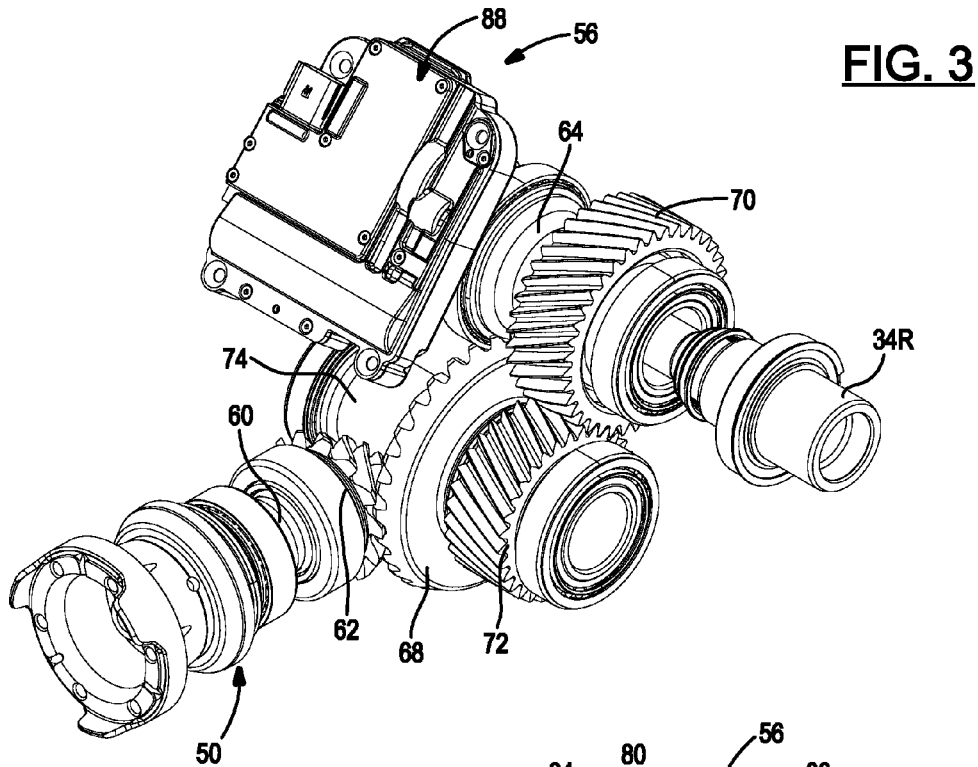


FIG. 5

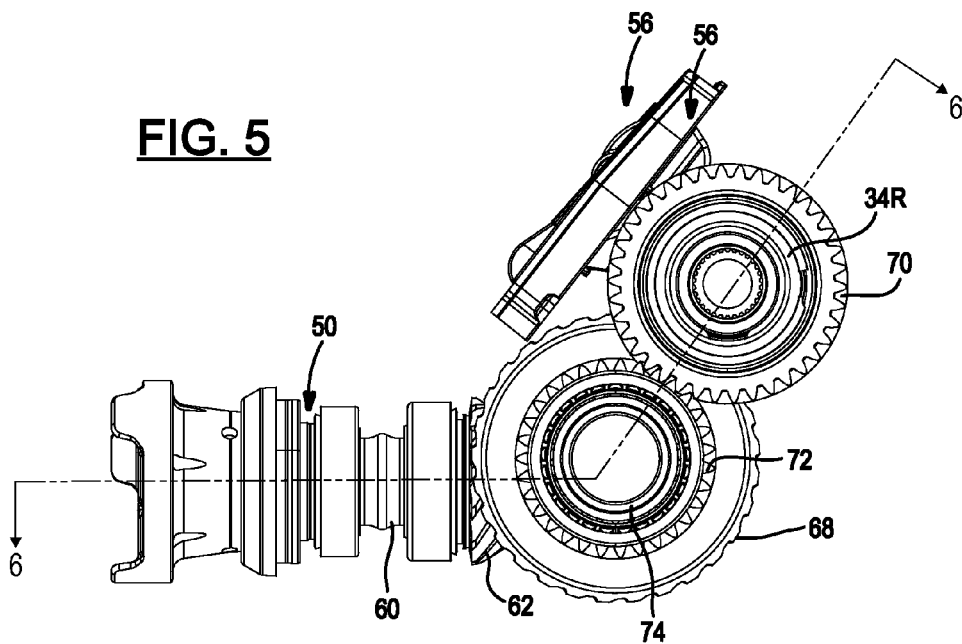
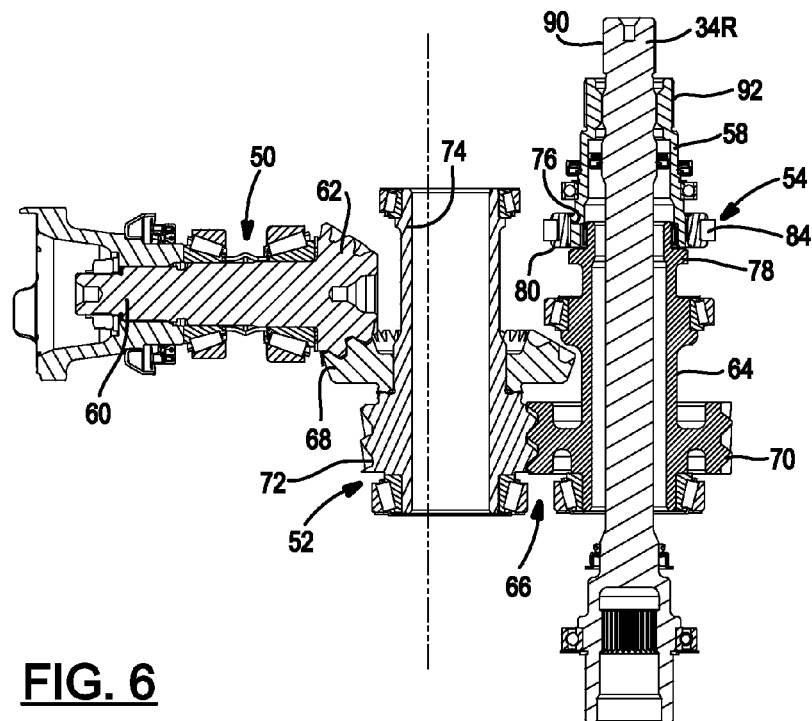
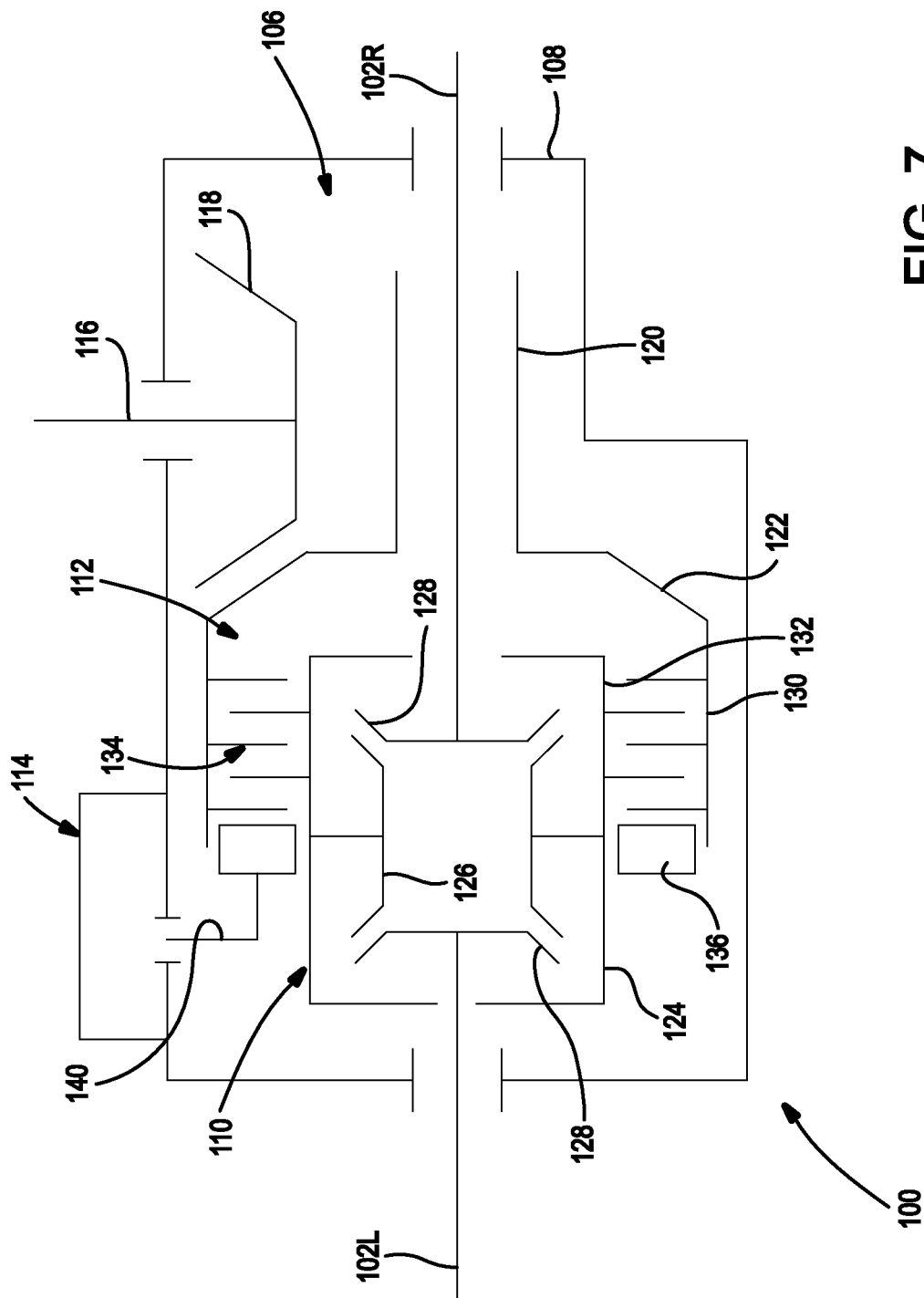


FIG. 6





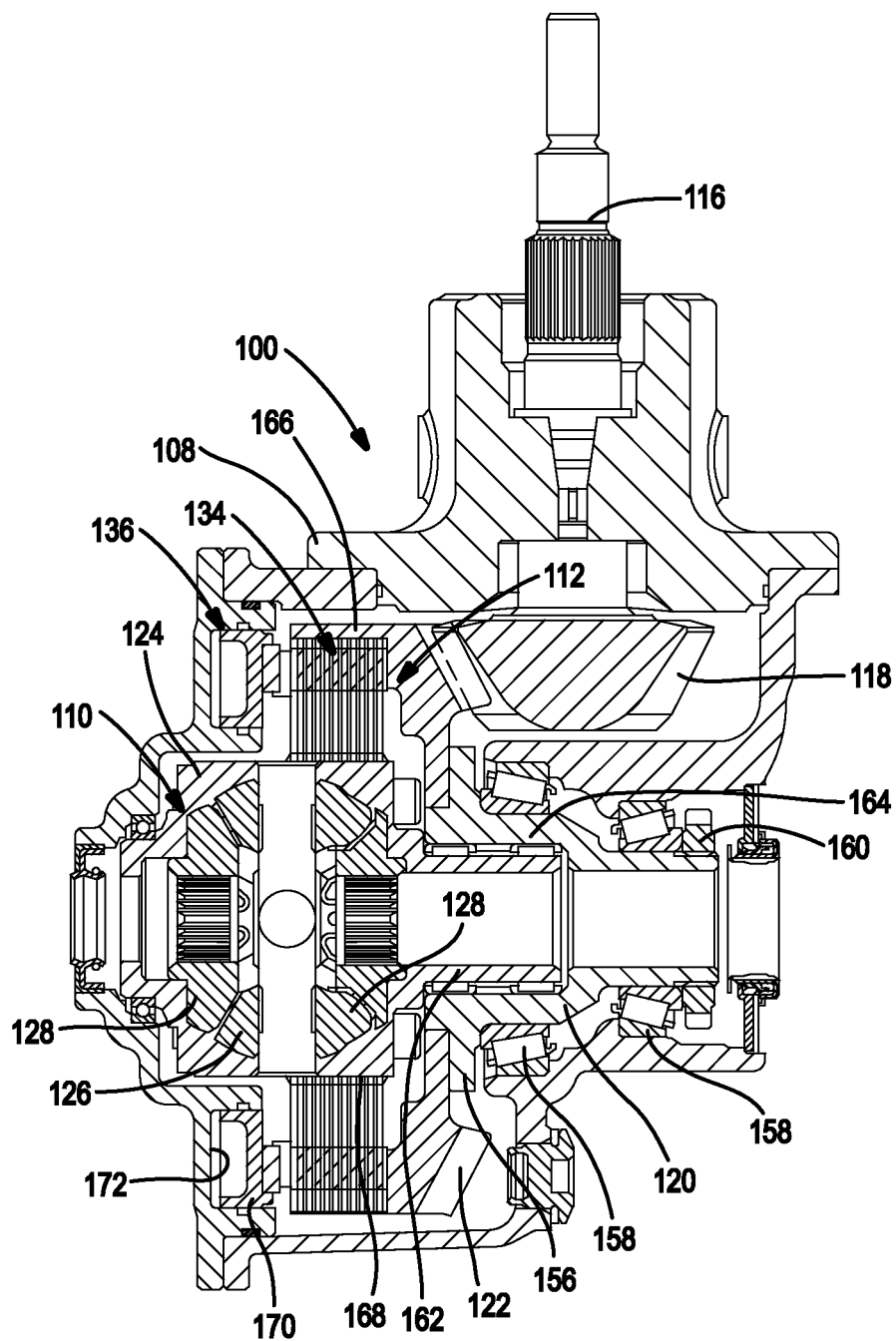


FIG. 8

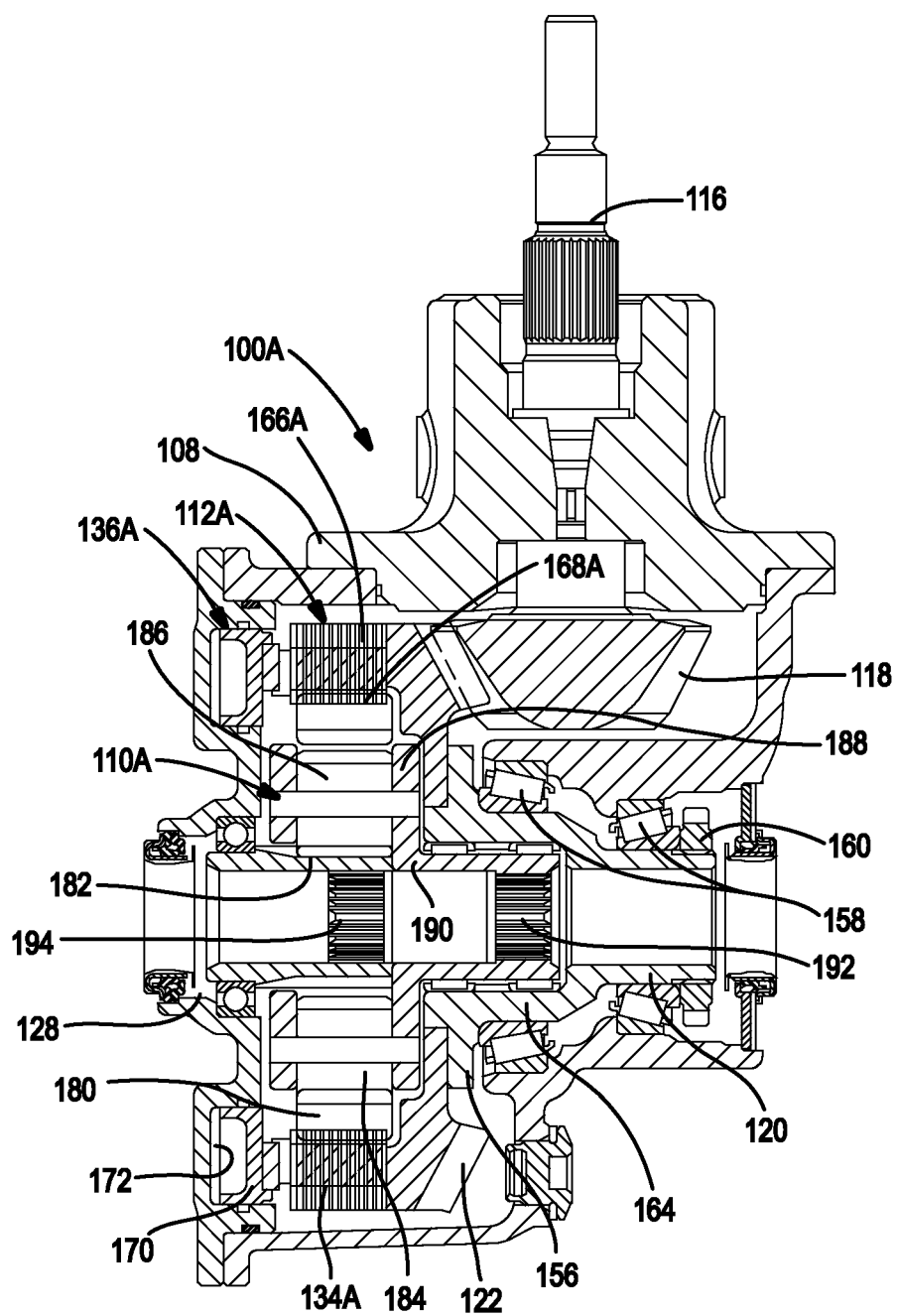


FIG. 9

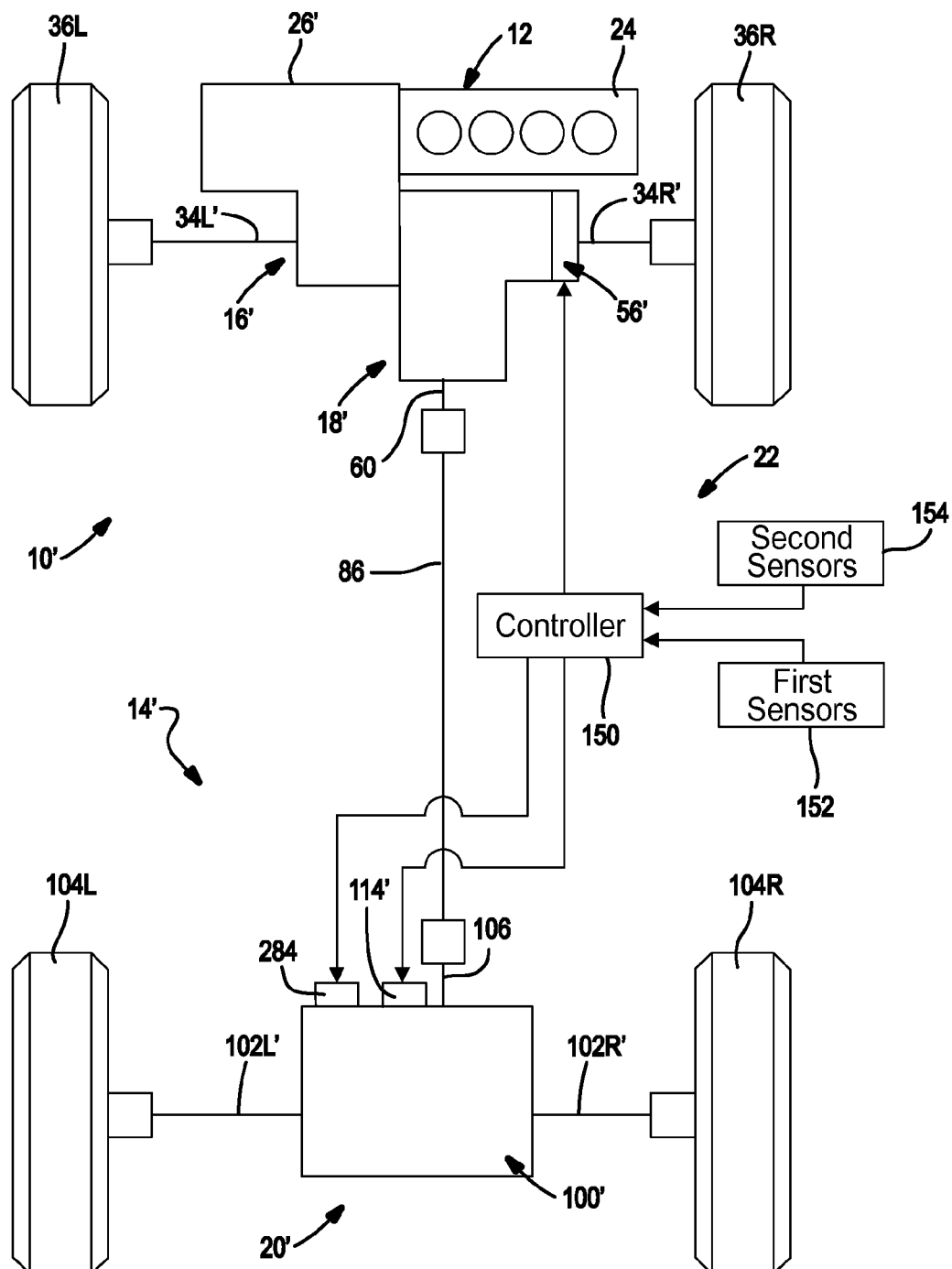


FIG. 10

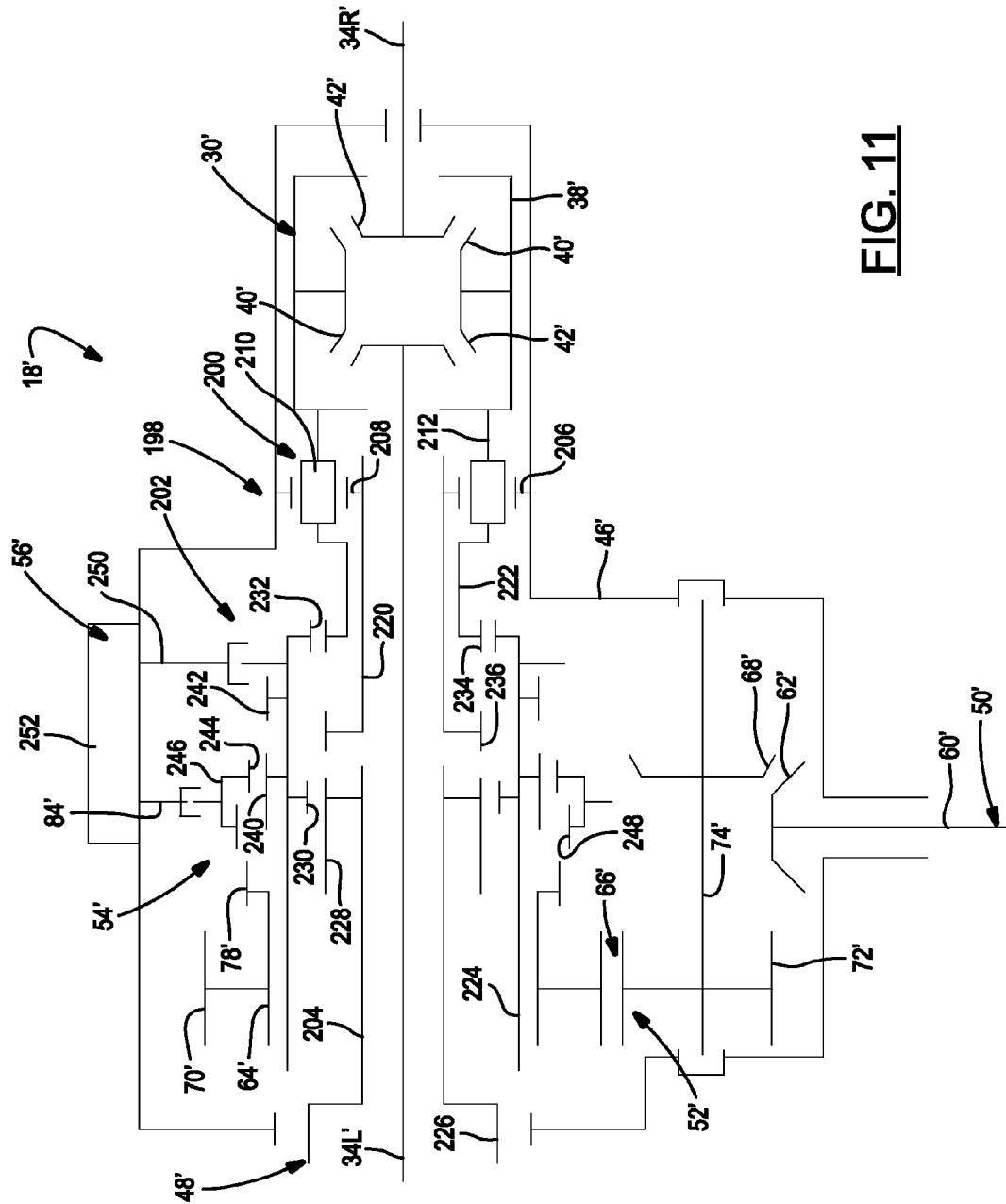


FIG. 11

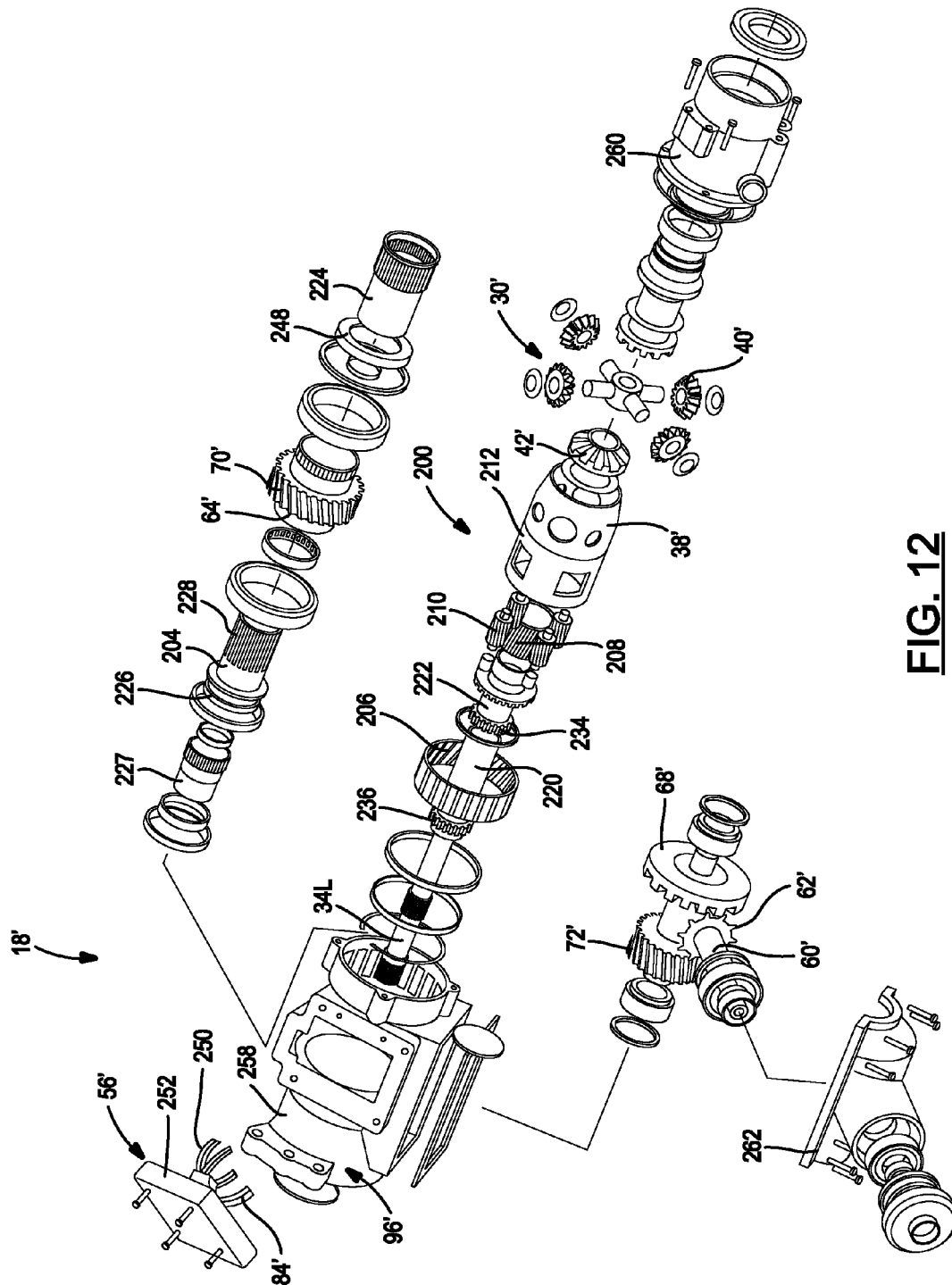


FIG. 12

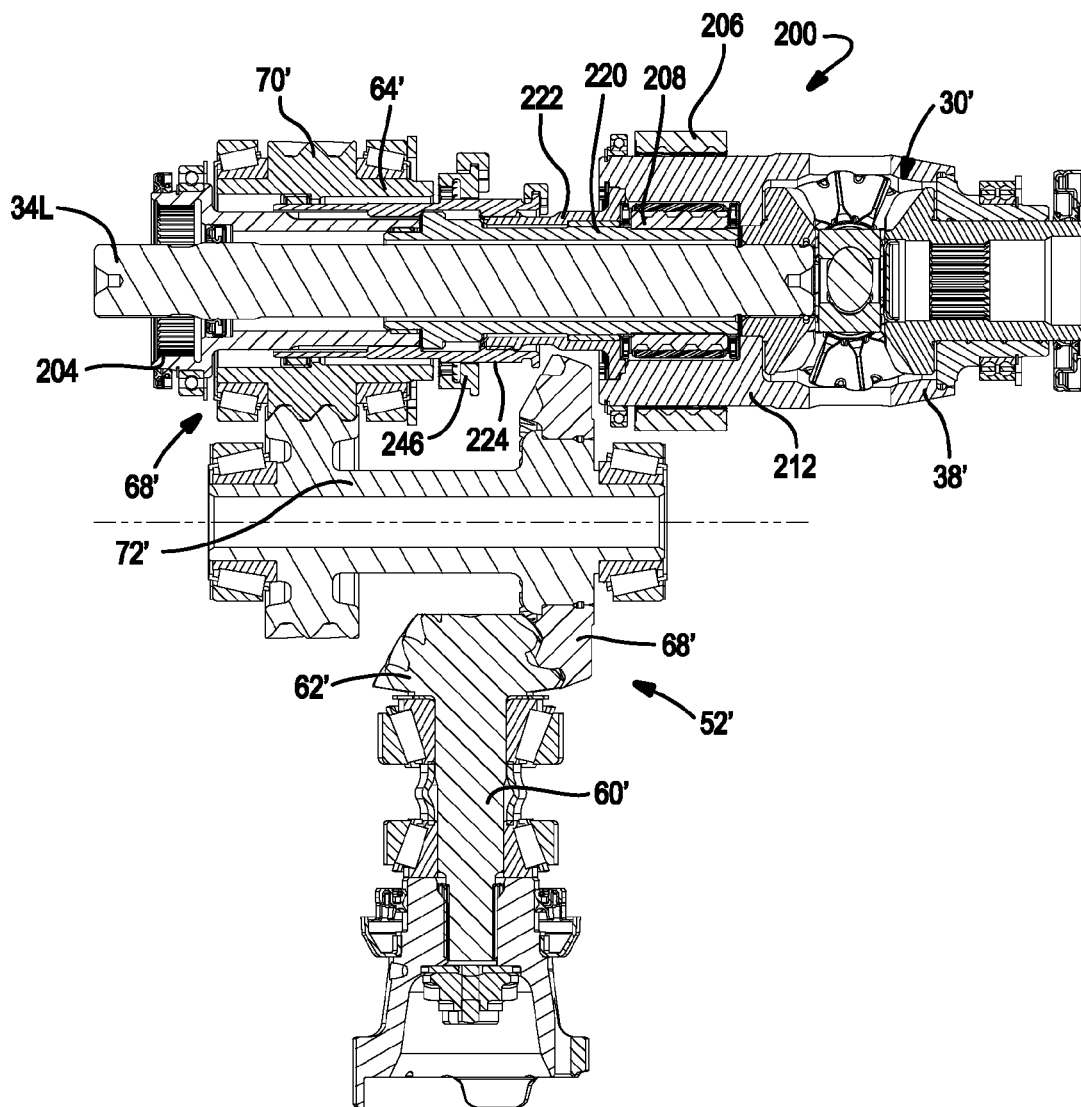


FIG. 13

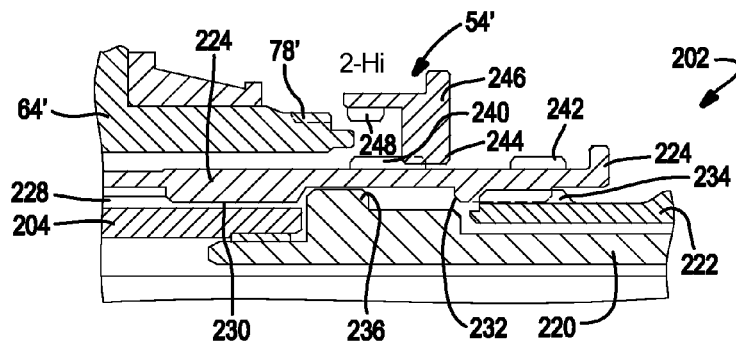


FIG. 14A

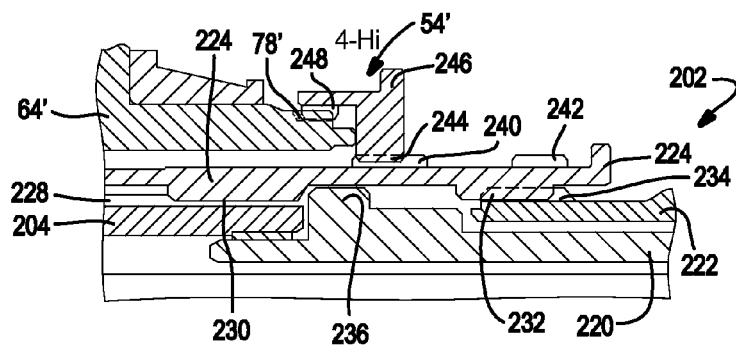


FIG. 14B

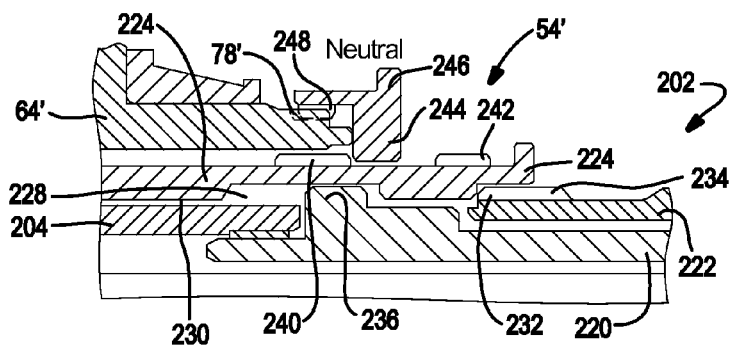


FIG. 14C

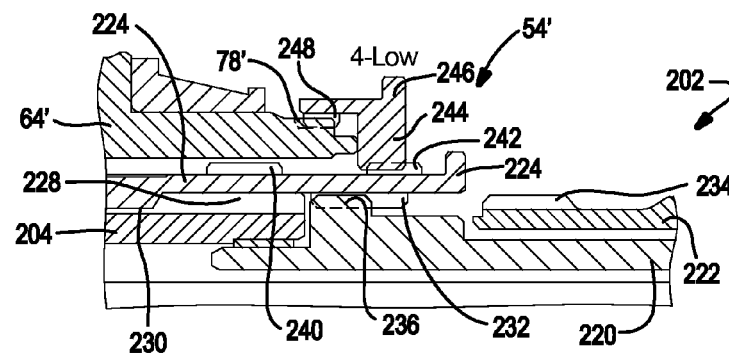


FIG. 14D

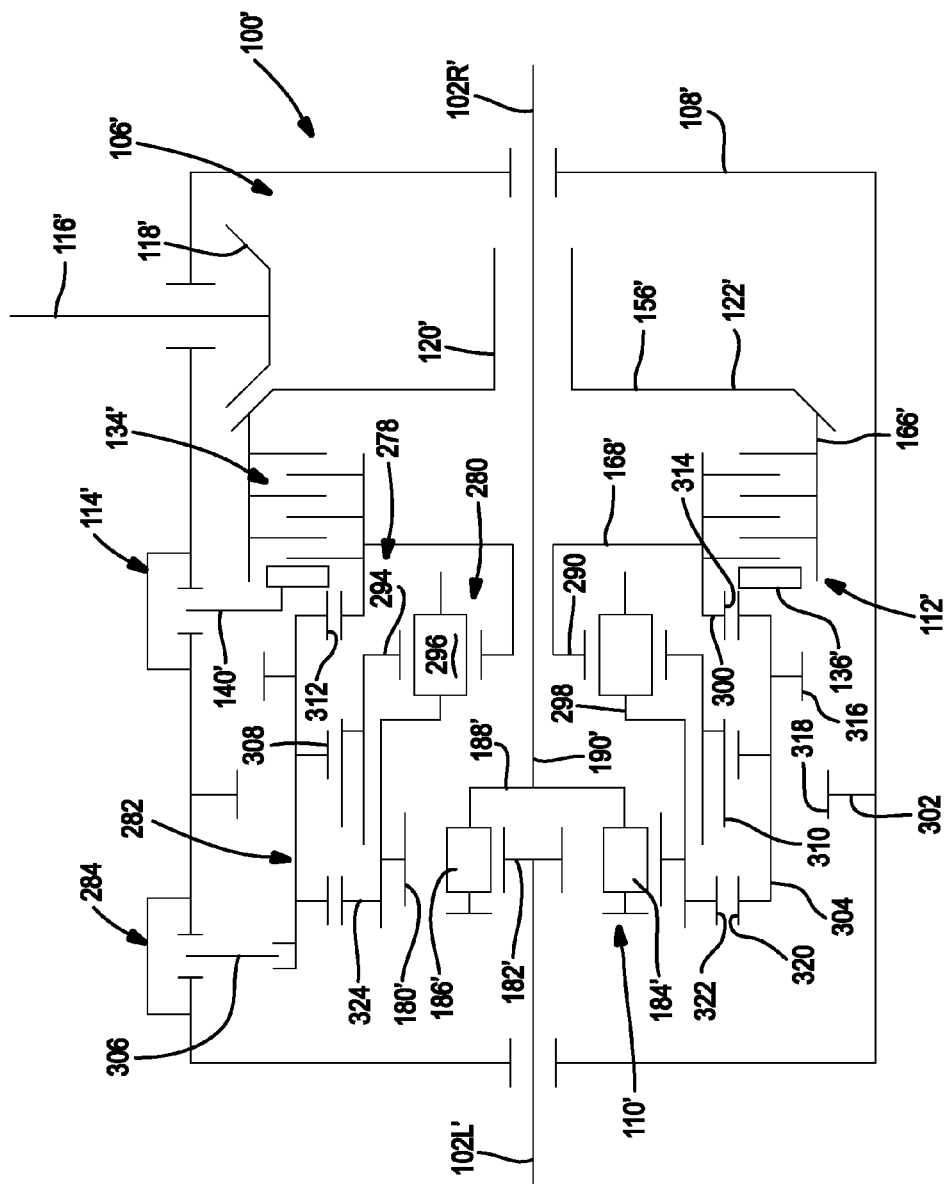
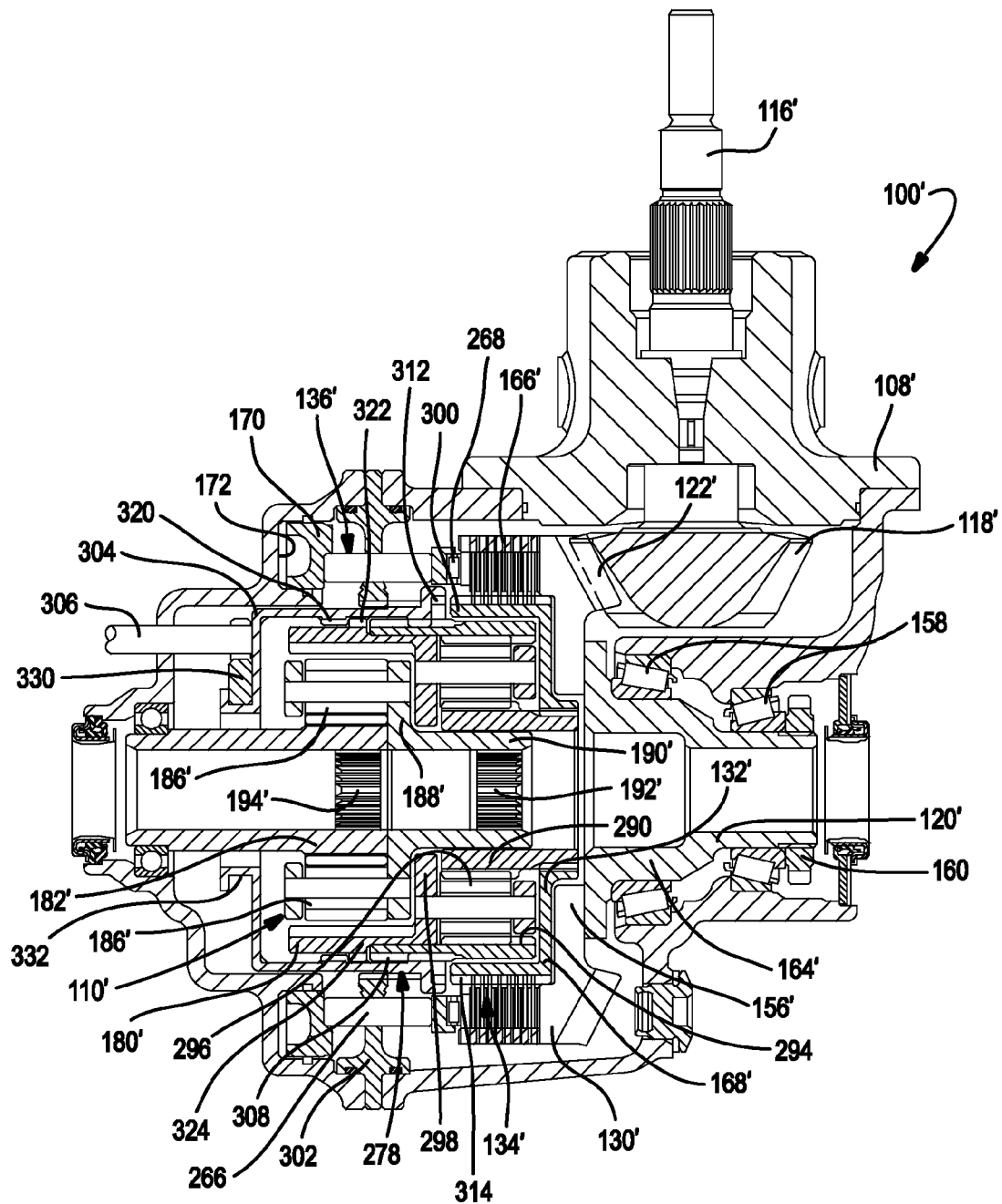


FIG. 15

FIG. 16A

**FIG. 16B**

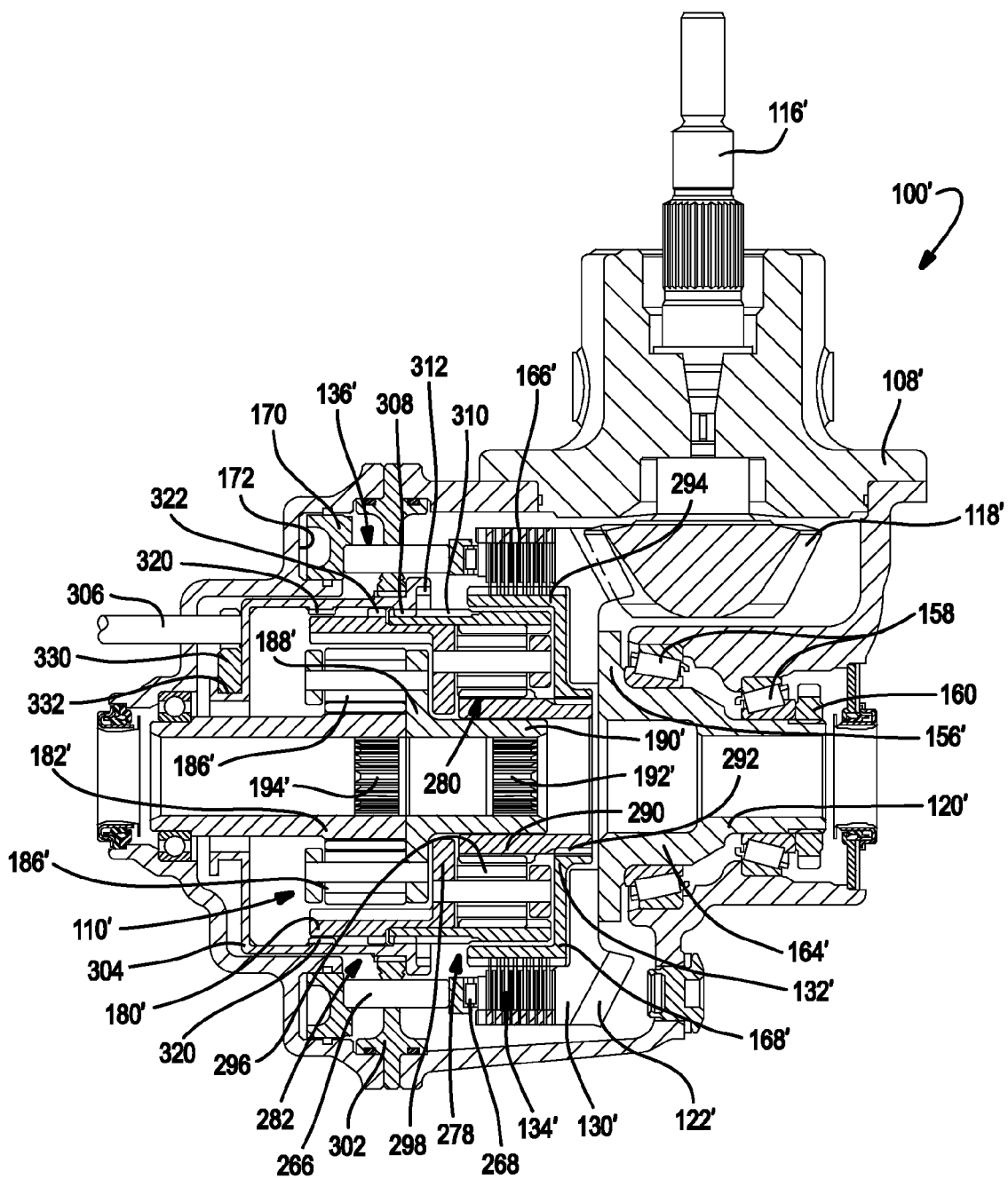


FIG. 16C

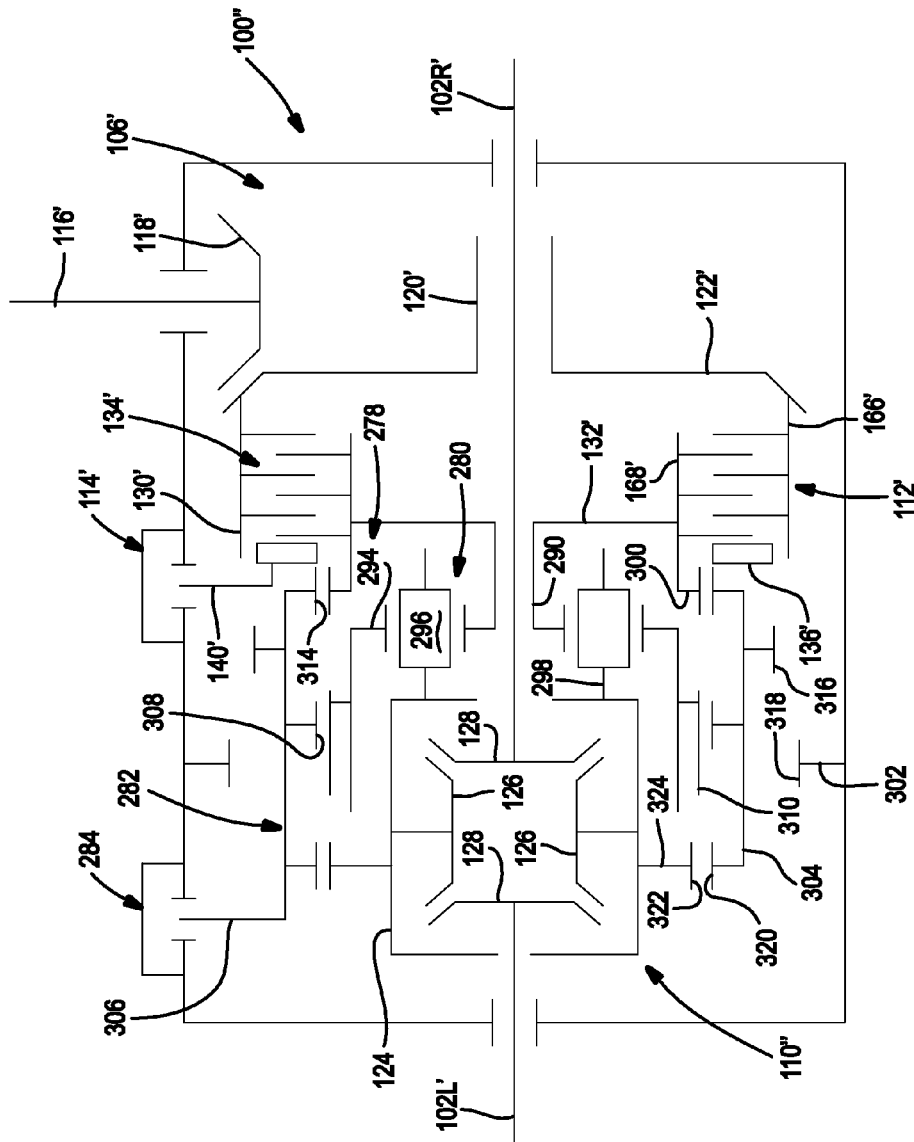


FIG. 17

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SINGLE SPEED AND TWO-SPEED DISCONNECTING AXLE ARRANGEMENTS

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation of U.S. patent application Ser. No. 14/239,756 filed Feb. 19, 2014, which is a U.S. National Phase application under 35 U.S.C. §371 of International Application No. PCT/US2013/063219, filed on Oct. 3, 2013, which claims the benefit and priority of U.S. 61/710,007, filed on Oct. 5, 2012. The entire disclosure of each of the above applications is incorporated herein by reference.

FIELD

The present disclosure relates to single speed and two-speed disconnecting axle arrangements.

BACKGROUND

This section provides background information related to the present disclosure which is not necessarily prior art.

Many modern automotive vehicles, such as crossover vehicles, are available with an all-wheel drive (AWD) drivetrain that is based on a front-wheel drive (FWD) architecture. This optional drivetrain arrangement permits drive torque to be selectively and/or automatically transferred from the powertrain to both the primary (i.e., front) driveline and the secondary (i.e., rear) driveline to provide better traction when the vehicle is operated in inclement weather and on off-highway road conditions. Such AWD vehicles necessarily are equipped with a much more complex drivetrain which, in addition to the primary driveline, must include the additional components associated with the secondary driveline such as a power take-off unit and a propshaft.

In an effort to minimize driveline losses (i.e., viscous drag, friction, inertia and oil churning) associated with secondary driveline being back-driven when no drive torque is transmitted thereto, it is known to incorporate a disconnect system that is configured to uncouple components of the secondary driveline such as, for example, the rear wheels or the rear differential from the remainder of the secondary driveline. To this end, there remains a need in the art for development of improved disconnectable drivelines for use in AWD vehicles.

SUMMARY

This section provides a general summary of the disclosure, and is not a comprehensive disclosure of its full scope or all of its features.

In one form, the present teachings provide a driveline component with a housing, a pinion mounted in the housing for rotation about a first axis, a ring gear and a pair of output members. The ring gear is meshed with the pinion and is rotatable about a second axis. The ring gear is supported for rotation relative to the housing by a pair of tapered roller bearings that are disposed along the second axis on a common lateral side of the ring gear. The output members are drivingly coupled to the ring gear.

In another form, the present teachings provide a drivetrain for an all-wheel drive motor vehicle. The drivetrain includes a primary driveline, a power switching mechanism and a second driveline. The primary driveline includes a first differential having a first differential case. The primary driveline is configured to drive a pair of first vehicle wheels. The power switching mechanism has an input shaft that is configured to

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receive rotary power from a powertrain and a disconnect mechanism that can be operated in a connected mode, which permits transmission of rotary power between the input shaft and the output pinion shaft, and a disconnected mode that inhibits transmission of rotary power between the input shaft and the output pinion shaft. The second driveline includes a propshaft and a rear drive module. The rear drive module includes a second differential, an input pinion shaft driving a hypoid gear and a torque transfer device. The second differential has an input member and a pair of output members that are configured to drive a pair of second vehicle wheels. The input pinion shaft is coupled by the propshaft to the output pinion shaft of the power switching mechanism. The torque transfer device is surrounds the planetary gear assembly and includes a clutch input member driven by the hypoid gear, a clutch output member driving the planetary gear assembly, and a clutch pack operably disposed between the input clutch member and the output clutch member. The torque transfer device is operable in a first mode, which permits transmission of rotary power from the clutch input member to the clutch output member, and a second mode that inhibits transmission of rotary power from the clutch input member to the clutch output member.

Further areas of applicability will become apparent from the description provided herein. The description and specific examples in this summary are intended for purposes of illustration only and are not intended to limit the scope of the present disclosure.

DRAWINGS

The drawings described herein are for illustrative purposes only of selected embodiments and not all possible implementations, and are not intended to limit the scope of the present disclosure.

FIG. 1 is a schematic of a motor vehicle equipped with a disconnectable all-wheel drive system constructed in accordance with the present teachings;

FIG. 2 is a schematic illustration of a single-speed power take-off unit associated with the disconnectable all-wheel drive system of FIG. 1;

FIG. 3 through FIG. 5 are perspective views of a single-speed power take-off unit based on the schematic shown in FIG. 2 with its housing structure removed for improved clarity and which is constructed in accordance with the present teachings;

FIG. 6 is a sectional view of the single-speed power take-off unit taken generally along line 6-6 of FIG. 5;

FIG. 7 is a schematic illustration of a single-speed rear drive module associated with the disconnectable all-wheel drive system of FIG. 1;

FIG. 8 is a sectional view of a single-speed rear drive module based on the schematic shown in FIG. 7 and which is constructed in accordance with the present teachings;

FIG. 9 is a sectional view of another single-speed rear drive module constructed in accordance with the present teachings and which can also be associated with the disconnectable all-wheel drive system of FIG. 1;

FIG. 10 is a schematic of a motor vehicle equipped with another configuration of a disconnectable all-wheel drive system constructed in accordance with the present teachings;

FIG. 11 is a schematic illustration of a two-speed power take-off unit associated with the disconnectable all-wheel drive system of FIG. 10;

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FIG. 12 is an exploded perspective view of a two-speed power take-off unit based on the schematic shown in FIG. 11 and which is constructed in accordance with the present teachings;

FIG. 13 is a sectional view of the two-speed power take-off unit shown in FIG. 11;

FIGS. 14A through 14D are partial sectional views of the two-speed power take-off unit shown in FIG. 13 with its mode and range shift components positioned to define a two-wheel high-range (2-Hi) mode, a four-wheel high-range (4-Hi) mode, a neutral mode, and a four-wheel low-range (4-Low) mode, respectively;

FIG. 15 is a schematic illustration of a two-speed rear drive module associated with the disconnectable all-wheel drive system of FIG. 10;

FIGS. 16A through 16C are sectional views of a two-speed rear drive module based on the schematic shown in FIG. 15 with its range shift components positioned to define a high-range (H) mode, a neutral (N) mode, and a low-range (L) mode; and

FIG. 17 is a schematic of another two-speed rear drive module associated with the disconnectable all-wheel drive system of FIG. 10.

Corresponding reference numerals indicate corresponding parts throughout the several views of the drawings.

DETAILED DESCRIPTION

The following exemplary embodiments are provided so that the present disclosure will be thorough and fully convey the scope to those skilled in the art. Numerous specific details are set forth such as examples of specific components, devices and schematic configurations to provide a thorough understanding of exemplary embodiments of the present disclosure. However, it will be apparent to those skilled in the art that these specific details need not be employed, that the exemplary embodiments may be embodied in many different forms, and that neither should be construed to limit the scope of the present disclosure.

With reference to FIG. 1 of the drawings, a motor vehicle constructed in accordance with the teachings of the present disclosure is schematically shown and generally indicated by reference numeral 10. The vehicle 10 can include a powertrain 12 and a drivetrain 14 that can include a primary driveline 16, a power switching mechanism 18, a secondary driveline 20, and a control system 22. In the various aspects of the present teachings, the primary driveline 16 can be a front driveline while the secondary driveline 20 can be a rear driveline.

The powertrain 12 can include a prime mover 24, such as an internal combustion engine or an electric motor, and a transmission 26 which can be any type of ratio-changing mechanism, such as a manual, automatic, or continuously variable transmission. The prime mover 24 is operable to provide rotary power to the primary driveline 16 and the power transfer mechanism 18.

With additional reference to FIG. 2, the primary driveline 16 can include a primary or first differential 30 having an input member 32 driven by an output member (not shown) of the transmission 26. In the particular construction shown, the first differential 30 is configured as part of the transmission 26, a type commonly referred to as a transaxle and typically used in front-wheel drive vehicles. The primary driveline 16 can further include a pair of first axle shafts 34L, 34R that can couple output components of the first differential 30 to a set of first vehicle wheels 36L, 36R. The first differential 30 can include a first differential case 38 that is rotatably driven by

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the input member 32, at least one pair of first pinion gears 40 rotatably driven by the first differential case 38, and a pair of first output side gears 42 meshed with the first pinion gears 40 and which are connected to drive the first axle shafts 34L, 34R.

With continued reference to FIG. 2, the power switching mechanism 18, hereinafter referred to as a power take-off unit (PTU), can generally include a housing 46, an input 48 coupled for common rotation with the first differential case 38 of the first differential 30, an output 50, a transfer gear assembly 52, a disconnect mechanism 54, and a disconnect actuator 56. The input 48 can include a tubular input shaft 58 rotatably supported by the housing 46 and which concentrically surrounds a portion of the first axle shaft 34R. A first end of the input shaft 58 can be coupled for rotation with the first differential case 38. The output 50 can include an output pinion shaft 60 rotatably supported by the housing 46 and having a pinion gear 62. The transfer gear assembly 52 can include a hollow transfer shaft 64, a helical gearset 66, and a hypoid gear 68 that is meshed with the pinion gear 62. The transfer shaft 64 concentrically surrounds a portion of the first axle shaft 34R and is rotatably supported by the housing 46. The helical gearset 66 can include a first helical gear 70 fixed for rotation with the transfer shaft 64 and a second helical gear 72 which is meshed with the first helical gear 70. The second helical gear 72 and the hypoid gear 68 are integrally formed on, or fixed for common rotation with, a stub shaft 74 that is rotatably supported in the housing 46.

The disconnect mechanism 54 can comprise any type of clutch, disconnect or coupling device that can be employed to selectively transmit rotary power from the powertrain 14 to the secondary driveline 20. In the particular example provided, the disconnect mechanism 54 is configured as a dog clutch. The dog clutch can include a set of external spline teeth 76 formed on a second end of the input shaft 58, a set of external clutch teeth 78 formed on the transfer shaft 64, a mode collar 80 having internal spline teeth 82 constantly meshed with the external spline teeth 76 on the input shaft 58, and a shift fork 84 operable to axially translate the shift collar 80 between a first mode position and a second mode position. While schematically shown as a non-synchronized dog clutch, it will be understood that the disconnect mechanism 54 can include a synchronized dog clutch if such a configuration is desired.

The mode collar 80 is shown in its first mode position, identified by a "2WD" leadline, wherein the internal spline teeth 82 on the mode collar 80 are disengaged from the external clutch teeth 78 on the transfer shaft 64. As such, the input shaft 58 is disconnected from driven engagement with the transfer shaft 64. Thus, no rotary power is transmitted from the powertrain 12 to the transfer gear assembly 52 and the output pinion shaft 60 of the power take-off unit 18. With the mode collar 80 in its second mode position, identified by an "AWD" leadline, its internal spline teeth 82 are engaged with both the external spline teeth 76 on the input shaft 58 and the external clutch teeth 78 on the transfer shaft 64. Accordingly, the mode collar 80 establishes a drive connection between the input shaft 58 and the transfer shaft 64 such that rotary power from the powertrain 12 is transmitted through the power take-off unit 18 to the output pinion shaft 60. As will be detailed, the output pinion shaft 60 is coupled via a propshaft 86 to the secondary driveline 20.

The disconnect actuator 56 can be any type of actuator mechanism that is operable for axially moving the shift fork 84 which, in turn, causes concurrent axial translation of the mode collar 80 between its two distinct mode positions. The disconnect actuator 56 is shown mounted to the housing 46 of

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the power take-off unit **18**. The disconnect actuator **56** can be a power-operated mechanism that can receive control signals from the control system **22** and can include, for example, hydraulically-actuated, pneumatically-actuated or electro-mechanically-actuated arrangements.

As noted, FIG. 2 schematically illustrates the components that can be associated with the power take-off unit **18**. Reference now to FIG. 3 through 6 will provide a more definitive structural configuration of such components that are associated with an exemplary embodiment of the power take-off unit **18**. In particular, these drawings illustrate the components in an assembled condition with the housing **46** removed for improved clarity. Each of the input shaft **58**, the transfer shaft **64**, the stub shaft **74**, and the output pinion shaft **60** are shown with suitable bearings assembled thereon for rotatably supporting each within or from the housing **46**. The disconnect actuator **56** is shown as a self-contained power-operated unit **88** from which the shift fork **84** extends. The power-operated unit **88** can include an electric motor and a geared drive unit configured to convert rotation of the motor output into translational movement of the shift fork **84**. External spline teeth **90** are provided on one end of the first axle shaft **34R** for facilitating a splined connection with its respective first side gear **42** in the first differential **30**. Likewise, external spline teeth **92** are provided on the first end of the input shaft **58** for facilitating a splined connection with a mating portion of the first differential case **38**.

With particular reference now to FIGS. 1 and 7, the secondary driveline **20** can include the propshaft **86**, a rear drive module (RDM) **100**, a pair of second axle shafts **102L**, **102R**, and a pair of second vehicle wheels **104L**, **104R**. A first end of the propshaft **86** can be coupled for rotation with the output pinion shaft **60** extending from the power take-off unit **18** while a second end of the propshaft **86** can be coupled for rotation with an input assembly **106** of the rear drive module **100**. The rear drive module **100** can include a housing **108**, a secondary or second differential **110**, a torque transfer device (TTD) **112** that is generally configured and arranged to selectively couple and transmit rotary power from the input assembly **106** to the second differential **110**, and a TTD actuator **114**. The input assembly **106** can include an input pinion shaft **116** having a pinion gear **118**, a hollow spool **120**, and a hypoid gear **122** fixed for rotation with the spool **120** and which is meshed with the pinion gear **118**. The second differential **110** can include an input member such as a second differential case **124**, at least one pair of second pinion gears **126** rotatably driven by the second differential case **124**, and a pair of output members such as second output side gears **128** that are meshed with the second pinion gears **126**. The second output side gears **128** are fixed for rotation with the inboard ends of the second axle shafts **102L**, **102R**. The second differential **110** and the torque transfer device **112** are shown disposed on one side of the input pinion shaft **116** to provide a compact arrangement.

The torque transfer device **112** can include any type of clutch or coupling device that can be employed to selectively transmit rotary power from the input assembly **106** to the second differential **110**. In the example shown, the torque transfer device **112** is a multi-plate friction clutch that can include an input clutch member **130** driven by the hypoid gear **122**, an output clutch member **132** coupled for rotation with the second differential case **124**, a multi-plate clutch pack **134** having a plurality of interleaved friction plates disposed between the input and output clutch members, and an engagement member **136** that is moveable for selectively applying a clutch engagement force to the clutch pack **134**. The torque transfer device **112** is shown to generally surround a portion

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of the second differential **110**. The TTD actuator **114** is provided to generate translational movement of the engagement member **136** relative to the clutch pack **134** and can be controlled by control signals from the control system **22**.

A first or “disconnected” mode can be established for the torque transfer device **112** when the engagement member **136** is positioned such that rotary power is not transmitted from the input clutch member **130** to the output clutch member **132**. In this “disconnected” mode, the second vehicle wheels **104L**, **104R**, the second axle shafts **102L**, **102R**, the second differential **110**, and the output clutch member **132** are disconnected from the input **106** of the rear drive module **100**. As such, rotation of these components as a result of rolling motion of the second vehicle wheels **104L**, **104R** does not “back-drive” input pinion shaft **116**, the propshaft **86** and components of the power take-off unit **18**.

A second or “connected” mode for the torque transfer device **112** can be established when the clutch engagement force exerted by the engagement member **136** on the clutch pack **134** causes rotary power to be transmitted from the input **106** to the second differential case **124** for delivery to the second vehicle wheels **104L**, **104R** through the second differential **110**. In addition, a “torque biasing” function can also be provided in the connected mode since variable control over the magnitude of the clutch engagement force applied to the clutch pack **134** can vary the distribution ratio of the rotary power transmitted from the powertrain **12** to the primary driveline **16** and the secondary driveline **20**. Thus, the torque transfer device **112** can be configured or controlled to slip or cyclically engage and disengage as appropriate for biasing the available drive torque while establishing the drive connection between the input **106** and the second differential **110**.

The TTD actuator **114** can be any power-operated device capable of shifting the torque transfer device **112** between its first and second modes as well as adaptively regulating the magnitude of the clutch engagement force exerted by the engagement member **136** on the clutch pack **134**. Thus, the TTD actuator **114** can, for example, include an electromagnetic or motor-driven ballscrew, ballramp or other cam actuation system having a mechanical connection, shown by lead line **140**, with the engagement member **136**. Alternatively, the TTD actuator **114** can include a hydraulic actuation system capable of regulating the position of the engagement member **136** relative to the clutch pack **134** by regulating fluid pressure, also indicated by lead line **140**, that is delivered to a pressure chamber.

The control system **22** is schematically shown in FIG. 1 to include a controller **150**, a group of first sensors **152**, and a group of second sensors **154**. The group of first sensors **152** can be arranged within the motor vehicle **10** to sense a vehicle parameter and responsively generate a first sensor signal. The vehicle parameter can be associated with any combination of the following: vehicle speed, yaw rate, steering angle, engine torque, wheel speeds, shaft speeds, lateral acceleration, longitudinal acceleration, throttle position and gear position without limitations thereto. The group of second sensors **154** can be configured to sense a driver-initiated input to one or more on-board devices and/or systems within the vehicle **10** and responsively generate a second sensor signal. For example, the motor vehicle **10** may be equipped with a sensor associated with a mode selection device, such as a switch associated with a push button or a lever, that senses when the vehicle operator has selected between vehicle operation in a two-wheel drive (FWD) mode and an all-wheel drive (AWD) mode. Also, switched actuation of vehicular systems such as the windshield wipers, the defroster, and/or the heating system, for example, may be used by the controller **150** to assess

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whether the motor vehicle **10** should be shifted automatically between the FWD and AWD modes.

As noted, FIG. 7 schematically illustrates the components that can be associated with the rear drive module **100**. Referring now to FIG. 8, a more definitive structural configuration of such components associated with an exemplary embodiment of the rear drive module **100** is shown. Specifically, the hypoid gear **122** can be fixed to a radial flange portion **156** of the bell-shaped spool **120** which, in turn, is rotatably supported from the housing **108** by a pair of laterally-spaced bearing assemblies **158**. A threaded nut **160** is installed on a threaded portion of the spool **120** and can be axially adjusted for varying the preload applied to the bearing assemblies **158**. The second differential case **124** has a hub extension **162** that can be rotatably supported within an enlarged bell portion **164** of the spool **120**. The input clutch member **130** can be operably associated with the hypoid gear **122** and can include a cylindrical clutch drum **166** that is fixed to or integrally formed with the hypoid gear **122**. The output clutch member **132** can be associated with the second differential case **124** and can include a clutch hub **168** that is fixed to or integrally formed on an outer surface of the second differential case **124**. A set of inner clutch plates of the clutch pack **134** can be splined to the clutch hub **168** while a set of outer clutch plates can be splined to the clutch drum **166**. As noted, the torque transfer device **112** is configured to surround a portion of the second differential **110** to provide a compact arrangement. The engagement member **136** can include an apply piston **170** assembly operably disposed for sliding movement in a pressure chamber **172** and which can be supplied with pressurized fluid by a hydraulically-operated unit associated with TTD actuator **114**.

Referring to FIG. 9, an alternative exemplary embodiment for the rear drive module **100** is shown and identified by reference numeral **100A**. The rear drive module **100A** is generally similar to rear drive module **100** but is equipped with an epicyclic-type second differential **110A** in place of the bevel-type second differential **110** shown in FIGS. 7 and 8. The epicyclic second differential **110A** can include an annulus gear **180**, a sun gear **182**, a set of first planet gears **184**, a set of the second planet gears **186**, and a carrier unit **188** from which the first planet gears **184** and second planet gears **186** are rotatably supported. The first planet gears **184** can be meshed with the annulus gear **180** while the second planet gears **186** can be meshed with the sun gear **182**. The first and second planet gears are circumferentially arranged such that each one of the first planet gears **184** also meshes with at least one of the second planet gears **186**. The annulus gear **180** acts as the input member for the second differential **110A** while the carrier unit **188** and the sun gear **182** act as the pair of output members. The carrier unit **188** can include a tubular boss **190** that is rotatably supported within the bell portion **164** of the spool **120** and which has a set of internal splines **192** configured to mate with a set of external splines (not shown) formed on an inboard end of the second axle shaft **102R**. Likewise, the sun gear **182** can have a set of internal splines **194** configured to mate with a set of external splines (not shown) formed on an inboard end of the second axle shaft **102L**.

The torque transfer device **112A** associated with the second differential **110A** can include a clutch drum **166A** fixed for rotation with the hypoid gear **122**, a clutch hub **168A** fixed for rotation with the annulus gear **180**, a clutch pack **134A** operably disposed therebetween, and a clutch engagement member **136A** operable to exert a clutch engagement force on the clutch pack **134A** in response to control signals transmitted by the control system **22** to the TTD actuator **114**. Thus,

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any rotary power transmitted by the input **106** through the clutch pack **134A** will drive the annulus gear **180** and be transmitted to the second axle shafts **102L**, **102R** through the sun gear **182** and the carrier unit **186**, respectively, while the meshed pairs of the first planet gears **184** and the second planet gears **186** facilitate speed differentiation between the second vehicle wheels **104L**, **104R**.

With reference to FIGS. 1, 2 and 7, the vehicle **10** can normally be operated in the two-wheel drive (FWD) mode in which the power take-off unit **18** and the rear drive module **100** are both disengaged. Specifically, the mode collar **80** of the disconnect mechanism **54** is positioned by the disconnect actuator **56** in its first mode position such that the input shaft **58** is uncoupled from the transfer shaft **64**. As such, substantially all power provided by the powertrain **12** is transmitted to the primary driveline **16**. Likewise, the torque transfer device **112** can be shifted into and maintained in its first mode such that the input **106**, the propshaft **86**, the output pinion shaft **60** and the transfer gear assembly **52** within the power take-off unit **18** are not back-driven due to rolling movement of the second vehicle wheels **104**.

When it is desired or necessary to operate the motor vehicle **10** in the all-wheel drive (AWD) mode, the control system **22** can be activated via a suitable input which, as noted, can include a drive requested input (via the mode select device) and/or an input generated by the controller **150** in response to signals from the first sensors **152** and/or the second sensors **154**. The controller **150** initially signals the TTD actuator **114** to shift the torque transfer device **112** into its second mode. Specifically, the controller **150** controls operation of the TTD actuator **114** such that the actuation member **136** is moved and a clutch engagement force is exerted on the clutch pack **134** that is sufficient to synchronize the speed of the secondary driveline **20** with the speed of the primary driveline **16**. Upon speed synchronization, the controller **150** signals the disconnect actuator **56** to cause the mode collar **80** in the power take-off unit **18** to move from its first mode position into its second mode position. With the mode collar **80** in its second mode position, rotary power is transmitted from the powertrain **12** to the primary driveline **16** and the secondary driveline **20**. It will be appreciated that subsequent control of the magnitude of the clutch engagement force generated by the torque transfer device **112** permits torque biasing across the clutch pack **134** for controlling the torque distribution ratio transmitted from the powertrain **12** to the primary driveline **16** and the secondary driveline **20**.

With reference to FIG. 10, another motor vehicle constructed in accordance with the present teachings is generally indicated by reference numeral **10'**. The vehicle **10'** is generally similar to the vehicle **10** of FIG. 1 except that the primary driveline **16'** and the secondary driveline **20'** associated with drivetrain **14'** have been modified to incorporate a two-speed range unit into both the power take-off unit **18'** and the rear drive module **100'**. As will be detailed, this alternative drivetrain arrangement for the vehicle **10'** permits establishment of at least one all-wheel low range drive mode in addition to the two-wheel high-range drive mode and the all-wheel high-range drive mode associated with vehicle **10**. For purposes of clarity, primed reference numeral are used to designate components that are generally similar in structure and/or function to the non-primed components previously described in relation to FIGS. 1 through 9.

With additional reference now to FIG. 11, the power take-off unit **18'** is generally shown to include a housing **46'**, an input **48'** adapted for connection to an output member of the transmission **26'**, an output **50'**, a transfer gear assembly **52'**, a first differential **30'**, a disconnect mechanism **54'**, a two-

speed range unit 198, and a disconnect actuator 56'. The input 48' can include a hollow input shaft 204 rotatably supported by the housing 46' and surrounding the first axle shaft 34L'. The output 50' can include an output pinion shaft 60' having a pinion gear 62'. The transfer gear assembly 52' can include a hollow transfer shaft 64', a helical gearset 66', and a hypoid gear 68' meshed with the pinion gear 62'. The helical gearset 66' can include a first helical gear 70' fixed for rotation with the transfer shaft 64' and a second helical gear 72' that is meshed with the first helical gear 70'. The second helical gear 72' and the hypoid gear 68' are integral with or fixed to a stub shaft 74' that is rotatably supported by the housing 46'.

The two-speed range unit 198 can include a planetary gear set 200 and a range shift device 202. The planetary gear set 200 can include a ring gear 206 non-rotatably fixed to the housing 46', a sun gear 208, a plurality of planet gears 210 meshed with both the ring gear 206 and the sun gear 208, and a planet carrier 212 from which the planet gears 210 are rotatably supported. The planet carrier 212 is fixed to, or integrally formed with, the first differential case 38' of the first differential 30' for common rotation therewith.

The range shift device 202 can include a sun gear shaft 220 surrounding a portion of the first axle shaft 34L' and which is fixed for rotation with the sun gear 208, a carrier shaft 222 surrounding a portion of the sun gear shaft 220 and which is fixed for rotation with the planet carrier 212, and a tubular range sleeve 224 surrounding portions of the carrier shaft 222, the sun gear shaft 220 and the input shaft 204. The input shaft 204 can have a first end 226 adapted for connection via a splined coupling shaft 227 (FIG. 12) to the output of transmission 26' and a second end having a set of elongated external spline teeth 228 formed thereon. The range sleeve 224 can include a set of internal spline teeth 230 that are in continuous meshed engagement with the external spline teeth 228 on the input shaft 204. As such, the range sleeve 224 is coupled for common rotation with the input shaft 204 while being capable of bi-directional axial sliding movement thereon between a plurality of predefined range position which will be discussed hereinafter in greater detail. The range sleeve 224 further defines a set of internal clutch teeth 232 that can be moved into and out of engagement with a set of external clutch teeth 234 formed on the carrier shaft 222 or a set of external clutch teeth 236 formed on the sun gear shaft 220.

The disconnect mechanism 54' is generally similar in function to the disconnect mechanism 54 in that it is configured to selectively connect the input shaft 204 to the transfer gear assembly 52' for transmitting rotary power from the input shaft 204 to the output pinion shaft 60' when the all-wheel drive mode is desired. However, the disconnect mechanism 54' differs in that the drive connection between the input shaft 204 and the transfer shaft 64' is made indirectly via the range sleeve 224. In particular, the range sleeve 224 can include first and second sets of external spline teeth 240 and 242, respectively, which are selectively engageable with internal spline teeth 244 formed on a mode collar 246. As such, the mode collar 246 is coupled for rotation with the range sleeve 224 and is capable of bi-directional axial translation relative to the range sleeve 224 between a first (2WD) mode position and a second (AWD) mode position.

With the mode collar 246 in its first mode position, a set of internal clutch teeth 248 formed on the mode collar 246 are released from meshed engagement with the external clutch teeth 78' on the transfer shaft 64', whereby no rotary power is transmitted from the input shaft 204 through the transfer gear assembly 52' to the output pinion shaft 60'. In contrast, with the mode collar 246 in its second mode position, its internal spline teeth 244 are engaged with one of the first and second

sets of external splines 240 and 242 (depending on the axial position of the range sleeve 224) and its internal clutch teeth 248 are engaged with the clutch teeth 78' on the transfer shaft 64', thereby establishing a drive connection between the input shaft 204 and the output pinion shaft 60'.

The two-speed range unit 198 is operable to establish at least two different speed ratio drive connections between the input shaft 204 and the first differential 30'. Specifically, the range sleeve 224 can be axially translated between a plurality of predefined range positions. In a first or "high" (Hi) range position, the range sleeve 224 is located such that its internal clutch teeth 232 are engaged with the external clutch teeth 234 on the carrier shaft 222. Since the internal splines 230 on the range sleeve 224 remain in constant meshed engagement with the external spline teeth 228 on the input shaft 204, location of the range sleeve 224 in its high-range position results in establishing a first or direct ratio drive connection between the input shaft 204 and the carrier shaft 222 which, in turn, is connected via the carrier 212 to the first differential case 38'. As such, a one-to-one or direct drive connection is established between the input shaft 204 and the first differential 30'.

In a second or "neutral" range position, the range sleeve 224 is disconnected from driven connection with both of the carrier shaft 222 and the sun gear shaft 220 such that the input shaft 204 is disconnected from the first differential 30'.

In a third or "low" (Low) range position, the range sleeve 224 is located such that its internal clutch teeth 232 are engaged with the external clutch teeth 236 formed on the sun gear shaft 220. With the range sleeve 224 located in its low-range position, a second or reduced-ratio drive connection is established between the input shaft 204 and the first differential 30'. Specifically, driven rotation of the sun gear shaft 220 causes the planetary gear set 200 to drive the carrier 212 at a reduced speed relative to the input shaft 204 such that the primary driveline 16' is likewise driven at the reduced speed ratio via the first differential 30'.

With continued reference to FIG. 11, the disconnect actuator 56' is shown positioned adjacent to the housing 46' and can include a first shift fork 84' engaging the mode collar 246, a second shift fork 250 engaging the range sleeve 224, and a power-operated unit 252 configured to receive control signals from the controller 150 and operable to coordinate movement of the shift forks 84' and 250. The power-operated unit 252 can be any type of unit capable of selectively translating the first shift fork 84' for causing movement of the mode collar 246 between its two mode positions while also selectively translating the second shift fork 250 for causing movement of the range sleeve 224 between its three range positions.

With reference now to FIGS. 12 and 13, a more definitive structural configuration of the components associated with the two-speed power take-off unit 18' is shown. In particular, FIG. 12 illustrates an exploded perspective view of an exemplary embodiment of the two-speed power take-off unit 18'. Housing 46' is shown to include a multi-piece assembly having a main housing 258 to which a differential housing 260 and a PTU housing 262 are secured. FIG. 13 is a sectional view which illustrates the compact arrangement of the planetary gear set 200, the range shift device 202, the transfer gear assembly 52', and the moveable mode collar 246 and range sleeve 224.

As will be understood, the bi-directional translational movement of the range sleeve 224 and the mode collar 246 can be coordinated to establish a plurality of range and mode combinations for the two-speed power take-off unit 18' based on control signals from the controller 150. Referring to FIGS.

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14A through 14D, these various range and mode combinations can be more clearly illustrated.

FIG. 14A shows the positions of the range sleeve 224 and the mode collar 246 for establishing a two-wheel high-range (2-Hi) mode for the power take-off unit 18'. Specifically, the mode collar 246 is shown located in its first mode position while the range sleeve 224 is shown located in its first range position. As such, the input shaft 204 is coupled via the range sleeve 224 to the carrier shaft 222 for establishing the direct drive connection between the powertrain 12 and the primary driveline 16'. Concurrently, the transfer shaft 64' is disconnected from driven connection with the input shaft 204, thereby disconnecting the secondary driveline 20' from the powertrain 12. Thus, rotary power is only transmitted by the powertrain 12 to the primary driveline 16' without speed reduction.

FIG. 14B shows the positions of the range sleeve 224 and the mode collar 246 for establishing a four-wheel high-range (4-Hi) mode for the power take-off unit 18'. Specifically, the high-range connection is maintained by the range sleeve 224 remaining in its first range position while the mode collar 246 is shown moved into its second mode position. Thus, the mode collar 246 establishes a drive connection from the input shaft 204 (through the range sleeve 224) to the transfer shaft 64' for also transmitting rotary power from the powertrain 12 to the secondary driveline 20'.

FIG. 14C shows the positions of the range sleeve 224 and the mode collar 246 for establishing a Neutral non-driven mode for the power take-off unit 18'. As seen, the mode collar 246 is maintained in its second mode position while the range sleeve 224 has been axially moved into its second range position such that its internal splines 232 are disengaged from the external clutch teeth 234 on the carrier shaft 220 and the external clutch teeth 236 on the sun gear shaft 220. Thus, the input shaft 204 is disconnected from both inputs to the primary driveline 16' such that no rotary power is transmitted from the powertrain 12 to the primary driveline 16'. It will also be noted that such movement of the range sleeve 224 to its second range position causes the internal spline teeth 244 on the mode collar 246 to disengage the first set of external splines 240 on the range sleeve 224 while the mode collar 246 maintains its connection with the transfer shaft 64'.

FIG. 14D shows the position of the mode collar 246 and the range sleeve 224 for establishing a four-wheel low-range (4-Low) mode for the power take-off unit 18'. Specifically, the mode collar 246 is maintained in its second mode position while the range sleeve 224 is moved axially into its third range position. As such, the low-range drive connection is established by the range sleeve 224 between the input shaft 204 and the sun gear shaft 220 while the AWD connection is established by the mode collar 246. It will be noted that the internal spline teeth 244 of the mode collar 246 engage the second set of external spline teeth 242 upon movement of the range sleeve 224 from its neutral range position into its low range position. While it is possible to provide the external splines 240 and 242 on the range sleeve 224 in a continuous arrangement, the non-toothed separation space therebetween has been recognized to inhibit potential tooth blocking conditions upon movement of the range sleeve 224 between its high-range and low-range positions.

With particular reference now to FIGS. 12, 15 and 16A through 16C, the secondary driveline 16' can include the propshaft 86, a two-speed rear drive module 100', a pair of second axle shafts 102L' and 102R', and a set of second vehicle wheels 104L and 104R. A first end of the propshaft 86 is coupled to the output pinion shaft 60' extending from the two-speed power take-off unit 18' while a second end of the

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propshaft 86 is coupled for rotation with an input assembly 106' of the two-speed rear drive module 100'. The rear drive module 100' can include a housing 108', a second differential 110', a torque transfer device 112', a TTD actuator 114' for controlling actuation of the torque transfer device 112', a two-speed range unit 278 having a planetary gear assembly 280 and a range shift mechanism 282, and a range actuator 284.

The input assembly 106' can include an input pinion shaft 116' having a pinion gear 118', a hollow spool 120', and a hypoid gear 122' fixed to a flange portion 156' of the spool 120' and which is meshed with the pinion gear 118'. The second differential 110' is an epicyclic arrangement which can include an annulus gear 180', sun gear 182', a set of first planet gears 184' meshed with the annulus gear 180', a set of second planet gears 186' meshed with the sun gear 182', and a carrier unit 188 from which the first planet gears 184' and the second planet gears 186' are rotatably supported. The planet gears are circumferentially arranged such that each one of the first planet gears 184' also meshes with at least one of the second planet gears 186'. The carrier unit 188' can include a tubular boss 190' that is configured to be connected via a splined connection 192' to the second axle shaft 102R' while the sun gear 182' can be connected via a splined connection 194' to the second axle shaft 102L'.

Torque transfer device 112' can include an input clutch member 130' fixed for rotation with the hypoid gear 122, an output clutch member 132', and a multi-plate clutch pack 134' operably disposed therebetween. A clutch drum 166' can be integrated with the hypoid gear 122' and act as the input clutch member 130' while a clutch hub 168' can act as the output clutch member 132'. The clutch pack 134' is operably disposed between the clutch drum 166' and the clutch hub 168'. The torque transfer device 112' can also include an engagement mechanism 136' that is moveable under the control of the TTD actuator 114' based on control signals from the controller 150 to selectively apply a clutch engagement force to the clutch pack 134'. Thus, rotary power transferred from the input assembly 106' through the torque transfer device 112' is transmitted to the clutch hub 168'. The engagement mechanism 136' can include an apply piston 170' disposed for sliding movement in a pressure chamber 172' which can be supplied with pressurized fluid by a hydraulically-operated unit associated with the TTD actuator 114'. The engagement mechanism 136' can further include a plurality of circumferentially aligned load pins 266 each having a first end engaging the apply piston 170' and a second end engaging an apply plate 268. The apply plate 268 is configured to apply the clutch engagement force on the clutch pack 134'.

The TTD actuator 114' can be any power-operated device capable of shifting the torque transfer device 112' between a first or "disconnected" mode and a second or "connected" mode. The first mode can be established when the engagement mechanism 136' is positioned such that rotary power is not transmitted from the input clutch member 130' to the output clutch member 132'. The second mode for the torque transfer device 112' can be established when the clutch engagement force exerted by the engagement mechanism 136' causes rotary power to be transmitted through the clutch pack 134' to the output clutch member 132'. The TTD actuator 114' can be generally similar to TTD actuator 114 and lead-line 140' is used to designate the mechanical or hydraulic connection between the TTD actuator 114' and the engagement mechanism 136'.

The two-speed range unit 278 is operable to establish at least two different speed ratio drive connections between the output clutch member 132' of the torque transfer device 112'

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and the second differential 110'. Specifically, the planetary gear assembly 280 can include a sun gear 290 fixed for rotation via a splined connection 292 with the clutch hub 168', a ring gear 294, a plurality of planet gears 296 meshed with the sun gear 290 and the ring gear 294, and a planet carrier 298 from which the planet gears 296 are rotatably supported. The planet carrier 298 can be fixed for common rotation with the annulus gear 180' of the second differential 110'.

The range shift mechanism 282 can include a first or direct clutch ring 300 fixed for rotation with the clutch hub 168', a second or low clutch ring 302 non-rotatably fixed to the housing 108', a range collar 304, and a range fork 306. The load pins 266 are shown in FIG. 16 to extend through a plurality of support bores formed through the second clutch ring 302. The range collar 304 can include a set of internal spline teeth 308 that is in continuous meshed engagement with a set of external spline teeth 310 formed on the ring gear 294 of the planetary gear assembly 280. As such, the range collar 304 is coupled for common rotation with the ring gear 294 while being capable of bi-directional axial sliding movement thereon. The range collar 304 can further include a set of first clutch teeth 312 that can be moved into and out of engagement with a set of clutch teeth 314 formed on the first clutch ring 300, a set of second clutch teeth 316 that can be moved into and out of engagement with a set of clutch teeth 318 formed on the second clutch ring 302, and a set of third clutch teeth 320 that can be moved into and out of engagement with a set of clutch teeth 322 formed on a third clutch ring 324 that is fixed for rotation with the annulus gear 180'. As will be detailed, translational movement of the range collar 304 is operable to establish at least two different speed range drive connections between the clutch hub 168' of the torque transfer device 112' and the annulus gear 180' of the second differential 110'.

The range collar 304 is shown in FIG. 16A positioned in a first or high range position such that its first clutch teeth 312 are meshed with the clutch teeth 314 on the first clutch ring 300 and its third clutch teeth 320 are meshed with the clutch teeth 322 on the annulus gear 180'. In addition, the second clutch teeth 316 on the range collar 304 are disengaged from engagement with the clutch teeth 318 on the second clutch ring 302. With the range collar 304 located in its first range position, the range collar 304 directly couples the clutch hub 168' to the annulus gear 180'. In addition, the sun gear 290 and the ring gear 294 are coupled together by the range collar 304 in its first range position such that the planetary gear assembly 280 is locked against relative rotation and rotates as a unit. Thus, the range collar 304 establishes a first or direct ratio drive connection between the output component (the output clutch member 132') of the torque transfer device 112' and the input component (the annulus gear 180') of the second differential 110' when located in its first range position.

The range collar 304 is shown in FIG. 16B positioned in a second or neutral range position such that its first clutch teeth 312 are disengaged from the clutch teeth 314 on the first clutch ring 300, its second clutch teeth 316 are disengaged from the clutch teeth 318 on the second clutch ring 302, and its third clutch teeth 320 are disengaged from the clutch teeth 322 on the third clutch ring 324. As such, the output clutch member 132' of the torque transfer device 112' is released from driven engagement with the annulus gear 180' of the second differential 110'.

The range collar 304 is shown in FIG. 16C positioned in a third or low range position such that its first clutch teeth 312 are disengaged from the clutch teeth 314 on the first clutch ring 300, its second clutch teeth 316 are meshed with the clutch teeth 318 on the second clutch ring 302, and its third

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clutch teeth 320 are disengaged from the clutch teeth 322 on the third clutch ring 324. As such, the ring gear 294 is braked against rotation and driven rotation of the sun gear 290 (via the output clutch member 132' of the torque transfer device 112') causes the planet carrier 298 to drive the annulus gear 180' at a reduced speed relative to the sun gear 290, thereby establishing a second or reduced ratio drive connection between the output clutch member 132' of the torque transfer device 112' and the annulus gear 180' of the second differential 110'.

The range actuator 284 can be any type of power-operated mechanism that is operable to control axial translational movement of the range fork 306 which, in turn, causes movement of the range collar 304 between its three distinct range positions. The range actuator 284 is shown schematically to be mounted to the housing 108' of the two-speed rear drive module 100'. The range actuator 284 can be a motor-driven geared device configured to receive control signals from the controller 150 and convert rotation of the motor output into translational movement of the range fork 306. The range fork 306 is shown in FIGS. 16A-16C to extend through an opening in the housing 108' and includes a projection 330 that extends into an annular groove 332 formed in the range collar 304.

In operation, the vehicle 10' can normally be operated in a two-wheel high-range drive mode in which the power take-off unit 18' establishes a high-range drive connection between the powertrain 12 and the primary driveline 16' while the rear drive module 100' is disengaged. Specifically, the range sleeve 224 and the mode collar 246 respectively associated with the range shift mechanism 202 and the disconnect mechanism 54' are located as shown in FIG. 14A to establish the 2-Hi mode. With the mode collar 246 in its first mode position, the input shaft 204 is disconnected from the transfer shaft 64' such that substantially all rotary power is transferred from the powertrain 12 to the primary driveline 16'. The torque transfer device 112' is maintained in its first mode to disconnect the secondary driveline 20'. While the torque transfer device 112' is operating in its first mode, the range collar 304 can be located in its high-range position (FIG. 16A).

When it is desired or necessary to operate the motor vehicle 10' in an all-wheel high-range (AWD-H) drive mode, the control system 22 can be activated to initially signal the TTD actuator 114' to shift the torque transfer device 112' into its second mode for synchronizing the speeds of the primary driveline 16' and the secondary driveline 20'. Upon synchronization, the controller 150 signals the disconnect actuator 56' to shift the mode collar 246 to its second mode position while maintaining the range sleeve 224 in its first range position (FIG. 14B). This establishes a four-wheel high-range drive connection between the powertrain 12, the primary driveline 16' and the input 106' to the rear drive module 100'. In addition, the range actuator 284 can be actuated to maintain or move the range collar 304 into its high-range position (FIG. 16A) such that the rotary power delivered through the torque transfer device 112' is transmitted to the second differential 110' at the direct speed ratio. Thereafter, the TTD actuator 114' can be controlled to vary the torque transmitted through the torque transfer device 112' to the second vehicle wheels 104L, 104R.

If during operation of the vehicle 10' in its AWD-H drive mode, it is desired or determined that improved traction requires operation in an all-wheel drive low-range (AWD-L) drive mode, the control system 22 functions to coordinate shifting of the power take-off unit 18' into its four-wheel low-range mode and the rear drive module 100' into its low-range mode. Specifically, the positions of the mode collar 246

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and the range sleeve **224** of the power take-off unit **18'** to establish this connection are shown in FIG. **14D** while the position of the range collar **304** of the rear drive module **100'** to establish this connection is shown in FIG. **16C**. Thus, the low-range drive connections are established in the power take-off unit **18'** and the rear drive module **100'**. These low-range drive connections can be established sequentially or concurrently based on a suitable control method and can be established with the vehicle **10'** in a stationary or non-motive state.

Finally, a towing mode for the vehicle **10'** can be established by shifting the power take-off unit **18'** into its neutral mode (FIG. **14C**) and the rear drive module **100'** into its neutral mode (FIG. **16B**).

Referring now to FIG. **17**, a schematic view of an alternative exemplary embodiment of a two-speed rear drive module **100''** is shown which can be associated with the vehicle **10'** shown in FIG. **10**. In essence, the two-speed rear drive module **100''** is generally similar to the two-speed rear drive module **100'** shown in FIGS. **15** and **16** with the exception that a bevel-type second differential **110''** is substituted for the epicyclic second differential **110'**. Thus, the two-speed range unit **278** is now operable to selectively establish drive connections between the output clutch member **132'** of the torque transfer device **112'** and the second differential case **124'** of the second differential **110''**. Specifically, movement of the range collar **304** between its three range positions is operable to establish the direct ratio, neutral and reduced-ratio drive connections between the output component of the torque transfer device **112'** and the input component to the second differential **110''**.

The present disclosure relates generally to single-speed and two-speed disconnectable drivelines for use in all-wheel drive vehicles. To this end, it is desirable to provide power take-off units and rear drive modules that are configured to provide a compact arrangement for modular assembly. The single-speed rear drive modules **100** and **100A** of FIGS. **8** and **9**, respectively, illustrates these desirable features in that the second differentials **110**, **110A** are shown installed in a cantilevered manner relative to the bell-shaped spool **120** and the meshed hypoid gear **122** and pinion gear **118**. The concentric location of the torque transfer devices **112**, **112A** relative to the second differentials **110**, **110A** provides a compact axial arrangement. In addition, the bevel-type differential **110** can be substituted for the epicyclic differential **110A** with only minor changes made to the housing **108** to permit product modularity. The two-speed rear drive modules **100'** and **100''** also show these desirable features and provide for a compact arrangement of the two-speed reduction units between the torque transfer devices and the second differentials.

While specific aspects have been described in the specification and illustrated in the drawings, it will be understood by those skilled in the art that various changes can be made and equivalents can be substituted for elements and components thereof without departing from the scope of the present teachings, as defined in the claims. Furthermore, the mixing and matching of features, elements, components and/or functions between various aspects of the present teachings are expressly contemplated herein so that one skilled in the art will appreciate from the present teachings that features, elements, components and/or functions of one aspect of the present teachings can be incorporated into another aspect, as appropriate, unless described otherwise above. Moreover, many modifications may be made to adapt a particular situation, configuration, or material to the present teachings without departing from the essential scope thereof. Therefore, it is intended that the present teachings not be limited to the par-

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ticular aspects illustrated by the drawings and described in the specification as the best mode presently contemplated for carrying out the present teachings, but that the scope of the present teachings include many aspects and examples following within the foregoing description and the appended claims.

What is claimed is:

1. A driveline component comprising:

a housing;

a pinion mounted in the housing for rotation about a first axis;

a ring gear meshed with the pinion and rotatable about a second axis, the ring gear being supported for rotation relative to the housing by a pair of tapered roller bearings that are disposed along the second axis on a common lateral side of the ring gear;

a pair of output members drivingly coupled to the ring gear; and

a differential assembly housed in the housing and drivingly disposed between the output members and the ring gear; wherein the differential assembly has a pair of differential outputs, each of the differential outputs being drivingly coupled to a corresponding one of the output members, the driveline component further comprising a clutch coupled to the differential assembly and selectively operable for disconnecting the pinion from the output members;

wherein the clutch is disposed between the ring gear and an input of the differential assembly; and

wherein the differential assembly comprises a planetary gearset and wherein an output of the clutch is coupled to a ring gear of the planetary gearset for common rotation.

2. A driveline component comprising:

a housing;

a pinion mounted in the housing for rotation about a first axis;

a ring gear meshed with the pinion and rotatable about a second axis, the ring gear being supported for rotation relative to the housing by a pair of tapered roller bearings that are disposed along the second axis on a common lateral side of the ring gear;

a pair of output members drivingly coupled to the ring gear; and

a differential assembly housed in the housing and drivingly disposed between the output members and the ring gear; wherein the differential assembly has a pair of differential outputs, each of the differential outputs being drivingly coupled to a corresponding one of the output members, the driveline component further comprising a clutch coupled to the differential assembly and selectively operable for disconnecting the pinion from the output members;

wherein the clutch is disposed between the ring gear and an input of the differential assembly; and

wherein the differential assembly comprises a pair of side gears that are housed in a differential case and wherein an output of the clutch is coupled to the differential case for common rotation.

3. The driveline component of claim 2, wherein the housing comprises a housing body and a pinion mount, the housing body defining a cavity into which the differential assembly is received, the pinion being rotatably mounted to the pinion mount, the pinion mount being removably coupled to the housing body.

4. The driveline component of claim 2, wherein the ring gear comprise a spindle portion onto which the tapered roller bearings are received, the spindle portion having a set of threads, and wherein a nut is threadably engaged to the set of

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threads and abutted to an inner bearing race of an outboard one of the tapered roller bearings.

5. A driveline component comprising:

a housing;

a pinion mounted in the housing for rotation about a first axis;

a ring gear meshed with the pinion and rotatable about a second axis, the ring gear being supported for rotation relative to the housing by a pair of tapered roller bearings that are disposed along the second axis on a common lateral side of the ring gear;

a pair of output members drivingly coupled to the ring gear; and

a differential assembly housed in the housing and drivingly disposed between the output members and the ring gear; wherein the differential assembly has a pair of differential outputs, each of the differential outputs being drivingly coupled to a corresponding one of the output members, the driveline component further comprising a clutch coupled to the differential assembly and selectively operable for disconnecting the pinion from the output members;

wherein the clutch is a friction clutch; and

wherein the housing comprises a housing body and a cover that is removably coupled to the housing body, wherein the differential assembly is received in a cavity in the housing body, and wherein a piston of the friction clutch is housed in an annular cavity defined by the cover.

6. The driveline component of claim 5, wherein a bearing is disposed between the cover and one of the differential outputs.

7. A drivetrain for an all-wheel drive motor vehicle, the drivetrain comprising:

a primary driveline including a first differential with a first differential case, the primary driveline being adapted for driving a pair of first vehicle wheels;

a power switching mechanism having an input shaft, which is adapted to receive rotary power from a powertrain, and a disconnect mechanism that is operable in a connected mode, which permits transmission of rotary power between the input shaft and an output pinion shaft, and a disconnected mode that inhibits transmission of rotary power between the input shaft and the output pinion shaft; and

a second driveline with a propshaft and a rear drive module, the rear drive module including a second differential, an input pinion shaft driving a hypoid gear, and a torque transfer device, the second differential having an input member and a pair of output members adapted to drive a pair of second vehicle wheels, the input pinion shaft being coupled by the propshaft to the output pinion shaft of the power switching mechanism, the torque transfer device surrounds the second differential and includes a clutch input member driven by the hypoid gear, a clutch

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output member driving the second differential, and a clutch pack operably disposed between the input clutch member and the output clutch member, the torque transfer device being operable in a first mode, which permits transmission of rotary power from the clutch input member to the clutch output member, and a second mode that inhibits transmission of rotary power from the clutch input member to the clutch output member.

8. The drivetrain of claim 7, wherein the power switching mechanism is a power take-off unit including a transfer gear assembly driving the output pinion shaft.

9. The drivetrain of claim 8, wherein the disconnect mechanism includes a mode collar that is movable between a first mode position and a second mode position, wherein the mode collar connects a differential case of the first differential to an input of the transfer gear assembly when the mode collar is in the first mode position, and wherein the mode collar is decoupled from one of the differential case and the input of the transfer gear assembly when the mode collar is in the second mode position.

10. The drivetrain of claim 7, wherein the rear drive module includes a housing and wherein the hypoid gear is supported for rotation relative to the housing by a pair of tapered roller bearings that are disposed along a rotational axis of the hypoid gear on a common lateral side of the hypoid gear.

11. The drivetrain of claim 10, wherein the housing comprises a housing body and a pinion mount, the housing body defining a cavity into which the second differential is received, the second pinion being rotatably mounted to the pinion mount, the pinion mount being removably coupled to the housing body.

12. The drivetrain of claim 10, wherein the hypoid gear comprise a spindle portion onto which the tapered roller bearings are received, the spindle portion having a set of threads, and wherein a nut is threadably engaged to the set of threads and abutted to an inner bearing race of an outboard one of the tapered roller bearings.

13. The drivetrain of claim 12, wherein a bearing is received in the spindle portion, the bearing supporting one of the output members of the second differential for rotation relative to the spindle portion.

14. The drivetrain of claim 7, wherein the second differential is an epicyclic differential assembly having an annulus gear, which serves as an input member of the epicyclic differential assembly, a sun gear, which serves as a first differential output of the epicyclic differential assembly, a carrier, which serves as a second differential output of the epicyclic differential assembly, and first and second planet gears rotatably supported from the carrier, wherein the first planet gear is meshed with the annulus gear and the second planet gear is meshed with the sun gear, and wherein the first planet gear is meshed with the second planet gear.

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